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MECHANICAL ENGINEERING

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After all, works of the engineer's creation are but the means to an end, not the end itself. They contribute to the well-being of humanity and man is more than a physical being. Surely, the engineer has a duty to perform in addition to the development and care of material things, wonderful as they are. I do not like to think that the engineer can see only the steel and masonry structure he designs and builds. I hope he has a vision of what it is built for as the architect of the Middle Ages had of the great cathedral he erected, and that the spiritual, moral, and esthetic sides of life have a great value to him.

ROBERT RIDGEWAY

President, American Society of Civil Engineers

OCTOBER 1925

THE MONTHLY JOURNAL PUBLISHED BY THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

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Publication Office, 207 Church Street, Easton, Pa. Editorial and Advertising Departments at the Headquarters of the Society, 29 West Thirty-ninth Street, New York

Volume 47 October, 1925 Number 10

CONTENTS OF THIS ISSUE

Power Plants for U. S. Navy Aircraft — *E. E. Wilson* — 783
Aeronautical Engineering at McCook Field — 790
- Achievements of American Aircraft-Engine Industry — *E. T. Jones* — 790
- Air-Cooled Engine Development — *S. D. Heron* — 791
- Generator Problems Due to Crankshaft Vibration — *J. W. Allen* — 793
- Fatigue of Welds — *R. R. Moore* — 794
- Earth-Induction Compass — *G. P. Luckey* — 795
- Magnesium and Its Alloys — *Samuel Daniels* — 796

A Review of Steam-Turbine Development — *Hans Dahlstrand* — 800
The Rational Design of Covering for Pipes Carrying Steam up to 800 Deg. Fahr — *W. A. Carter and E. T. Cope* — 805
Lake Waters for Condensers — *A. G. Christie* — 809
Comparison of Actual Performance and Theoretical Possibilities of the Lake-side Station — *M. K. Drewry* — 811
Steam Logging — *J. W. Gill* — 815
Electric Logging — *P. A. Wickes* — 820
Testing the Fineness of Pulverized Coal — 822
The Modern City and the Engineer's Relation to It — *Robert Ridgeway* — 824
Mechanical Features Affecting the Reliable and Economical Operation of Hydroelectric Plants — *E. A. Dow* — 825
Principles of Metallurgy of Ferrous Metals for Mechanical Engineers—VI — *Leon Cammen* — 832
Pumping-Engine Tests and Sewage Disposal at Milwaukee — 837
- Activated-Sludge Sewage-Disposal Plant at Milwaukee — *J. A. Wilson* — 837
- Duty Tests of Vertical Triple-Expansion Pumping Engines, Milwaukee, Wis — *C. A. Cahill* — 841

Tests of a Uniflow Engine — 844

DEPARTMENTAL

Survey of Engineering Progress — 846
New Data and Tendencies in the Technology of Firing; Short Abstracts of the Month

Engineering and Industrial Standardization — 858

Correspondence — 859
Weights of Castings on Blueprints; Factory Administration in India; How Much Can Be Gained by Exhaust Turbines?

Editorial Notes — 862
The Beginning of American Air Transport; Frontiers but no Boundaries; Supersaturation; Intelligent Wood Conservation; Tidal Power; The "Shenandoah;" Recognizing the Exceptional Student; An American Museum of Engineering; Industrial Engineering at Ohio State; Course in Military Engineering at M.I.T.; The New Haven Machine-Tool Exhibition

Library Notes and Book Reviews — 868

The Engineering Index — 127-EI — Late Items — 870

ADVERTISING

Display Advertisements — 1
Professional Engineering Service Section — 120
Opportunity Advertisements — 125
Classified List of Mechanical Equipment — 128
Alphabetical List of Advertisers — 148

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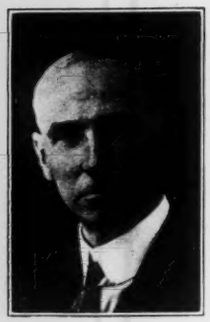
Entered as second-class matter at the Post Office at Easton, Pa., under the Act of March 3, 1879.
Acceptance for mailing at special rate of postage provided for in section 1103, Act of October 3, 1917, authorized on January 17, 1921.

P. A. WICKES

J. A. WILSON



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* * * * *

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WASTES IN DISTRIBUTION

Management Week, October 19–24, 1925

Wastes in Distribution will be the topic of the meetings to be held throughout the country during Management Week, the plans for which are being arranged by a Joint Committee of which Herbert Hoover is Honorary Chairman and which is made up of representatives of The American Management Association, The American Society of Mechanical Engineers, The National Association of Cost Accountants, The Society of Industrial Engineers, and The Taylor Society.

MECHANICAL ENGINEERING

Volume 47 October, 1925 No. 10

Power Plants for U. S. Navy Aircraft

A Résumé of Development, an Analysis of the Present Situation, and an Estimate of the Future

By LIEUT-COMDR. E. E. WILSON, U. S. N.,[1] WASHINGTON, D. C.

THE heart of the airplane is its power plant. From the beginning of man's efforts to fly he has needed only a suitable means of propulsion to make mechanical flight possible. Once he has succeeded in flight, he can continue to fly only so long as he has means of propulsion. The aircraft, then, is no better than its power plant.

The history of aircraft-engine development is written in the words "pounds per horsepower." Obviously, this ratio must be kept at a minimum, and this result can be attained by decreasing weight and increasing power. The specific weight is reflected back into the gross weight of the airplane. The heavier the engine for a given power, the greater the wing area necessary to support it. Larger wing area results in greater weight of the structure which, in turn, requires more power to attain the given performance. These are the controlling factors which have forced us steadily from a dry-engine weight of from 5 or 6 lb. per hp. to the modern weight of 1.4 lb. per hp., and from an engine power of 30 to 35 to one of 800.

The economy of a power plant is of importance not because of the cost of fuel but because of this same factor of weight. A good average specific fuel consumption for aircraft engines is 0.50 lb. per hp-hr. If a power-plant weight of about 3 lb. per hp. is assumed, which includes the engine and all the accessories, tanks, and piping necessary to fly, it is apparent that the modern engine will burn its own weight of fuel in six hours' cruising. Any improvement in economy results in a corresponding reduction of fuel weight for a given cruising range. This weight is again reflected in the weight of the structure and thus the economy of the power plant becomes a primary factor in the ratio pounds per horsepower.

Early Types of Power Plants

With this viewpoint firmly established, it now becomes desirable to survey the power-plant situation. Such a survey will include a brief résumé of the history of development, an analysis of present attainments, and an estimate of the future. One of the first successful aircraft engines was the Wright four-cylinder-in-line, 30 to 35-hp. engine. It was water-cooled and had a specific weight of $5^1/_2$ lb. per hp. which was too great. For a time aeronautic development was retarded by this limitation, but before long the first air-cooled rotary engines were developed. The crankshaft of these engines was fixed and the cylinders rotated. Two examples of this early type—the Gnome and LeRhone—weighed about $3^1/_2$ lb. per hp. and resulted in great progress in aircraft.

The two major inherent difficulties encountered in these engines are the fan resistance of the rotating cylinders and their gyroscopic effect. When, in the desire to increase power in these engines, their speed of rotation was increased, the fan work mounted so rapidly as definitely to limit their rotative speed. These factors resulted in the return to the water-cooled type, of which the Curtiss OX is a good example. This engine, developing 90 to 100 hp. and weighing about 4 lb. per hp., dry, is an 8-cylinder, 90-deg., V-type, the cylinders being staggered so that two connecting rods may run side by side on one crankpin.

Close on the heels of the OX followed the Hispano-Suiza, developing 150 hp. This is a water-cooled, 8-cylinder, 90-deg., V-type, weighing, dry, about 3.2 lb. per hp. About this time, came the demand for more power, and, with the entry of the United States into the World War, the Liberty engine project. It was originally intended to build this engine in three types—the 6-cylinder-in-line, the 8-cylinder V-type, and the 12 cylinder V-type. By the time the engine was developed, however, the necessity for the smaller types had disappeared and the Liberty, in its final form, is a 12-cylinder 45-deg. engine developing about 400 hp. and weighing at that power about 2.2 lb. per b.hp.

Here is a steady increase in power and a steady decrease in weight per horsepower. The vision which prompted the Liberty program, the rapidity with which it was accomplished, and the success that attended it after some minor difficulties at the start, are outstanding achievements in aeronautical engineering. The Liberty engine in use today is considerably modified. Crankshafts have been made stiffer and gears have been made stronger to withstand the torsional vibrations. Cylinders have been reinforced to prevent water leaks. The ignition has been modified from the 8-volt generator type to the 12-volt, and distributors are being converted from the brush type to the jump-spark type. The Liberty is still a good engine, but it has certain definite limitations which have made the development of new engines necessary.

Engine Development Since the War

Engine development stagnated for a time after the close of the war because of the large stock of engines on hand. Work was resumed shortly, however, and the first result was the Curtiss D-12, developing 325 to 410 hp. on a dry weight of 698 lb. and weighing, at 400 hp., 1.75 lb. per b.hp., dry. This engine is in use at the present time in all the high-speed and pursuit airplanes and was used in Lieutenant Maughan's famous dawn-to-dusk flight.

The advent of the torpedo airplane created an insistent demand for greater power. This resulted in the Wright type T, which was originally designed as a 500-hp. engine and is now being purchased as a 600-hp. engine weighing, dry, about 1180 lb., which, at its maximum power of 650, makes its specific weight about 1.8 lb. This engine has been and still is in quantity production and has proved itself to be very dependable.

As was indicated before, the Liberty is restricted by inherent characteristics to speeds of about 1700 r.p.m. Its power curve peaks at 1900, and its life between overhauls is seriously decreased if operated at crankshaft speeds above 1700. This resulted in the development of a new engine in this particular field, and the result was the Packard 1A-1500 in the 400 to 500-hp. class. This engine has been described in detail in a late issue[1] of the *Journal* of the Society of Automotive Engineers. It weighs 735 lb. dry and at a rating of 500 hp. at 2000 r.p.m. its specific weight becomes 1.47 lb. This engine is now in its limited production stage and, so far, has given excellent performance in the air.

The demand for a single-engine bomber of high power with heavy-load-carrying characteristics has resulted in the development of the new Packard 1A-2500 engine which has just lately completed its type tests, developing 834 hp. at 2050 r.p.m. On a dry weight of 1135 lb. its specific weight becomes 1.36. This engine has also

[1] Head of Engine Section, Bureau of Aeronautics, Navy Department.

Paper read at a meeting of the Central Pennsylvania Section of The American Society of Mechanical Engineers, Lewisburg, Pa., April 18, 1925.

[1] Recent Developments in Aircraft Engines, L. M. Woolson, *Journal*, Society of Automotive Engineers, vol. 16, no. 3, March, 1925, p. 297.

gone into limited production, in some cases with a gear ratio of 2 to 1, and it is expected to have it in the air within a few months. With gears, complete, the weight of this engine is 1300 lb.

AIR-COOLED ENGINES

The previous outline covers the water-cooled field. Air-cooled development has been much less rapid owing to certain inherent difficulties, but simultaneously with this water-cooled development there has been produced the 9-cylinder Wright model J, a static radial type developing 200 hp. on a weight of about 470 lb. This makes the engine weigh, dry, 2.35 lb. per hp., almost exactly that of the Wright model E-4 water-cooled engine of similar power. The E-4 is the Hispano engine improved and refined to increase its power, economy, and durability. This comparison of the air-cooled and water-cooled engines on a dry-weight basis emphasizes the necessity for comparing engines not on the dry or wet basis, but rather in the condition "ready to fly." Comparing these engines on their power-plant weights, it is found, as indicated in Fig. 1, that the relatively undeveloped air-cooled engine weighs about three-fourths as much as the very best of the water-cooled type in the same power.

In the 400-hp. air-cooled field is the Wright model P-1, which, at the very conservative rating of 400 hp., weighs about 800 lb. and has a specific weight, ready to fly, of about 2.3 lb. per hp. It is interesting to contrast this with the Liberty power-plant weight of about 3.3 lb. per b.hp., the difference in weight being

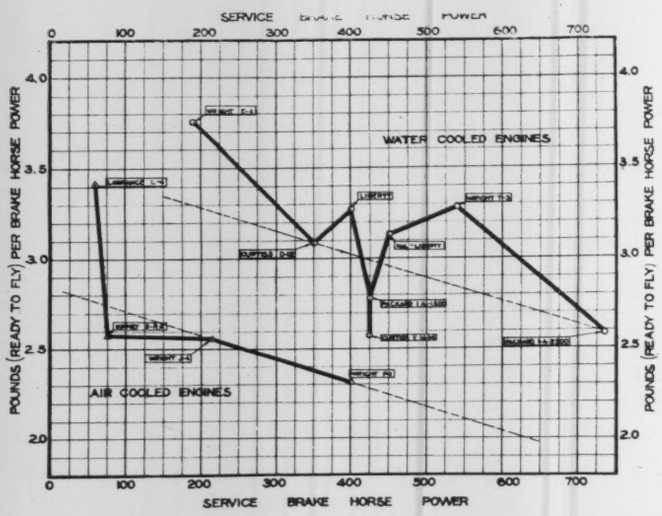

FIG. 1 COMPARISON OF WEIGHTS PER HORSEPOWER OF AIR-COOLED AND WATER-COOLED U. S. NAVY AIRCRAFT ENGINES ON A READY-TO-FLY BASIS

due to the weight of the radiator and the cooling system. Both the J and P models are static radials; that is, fixed-cylinder engines as contrasted to the early radials of the rotary type. The model P engine has lately passed its fifty-hour type tests very successfully and has given surprising performance in recent flight tests. This engine should be capable of delivering at least 475 hp. in its next form.

Summarizing the development and referring to the graph, Fig. 2, there is now a water-cooled engine in every power from 150 to 800, and air-cooled engines in the 200- to 400-hp. class. A 1000-hp. water-cooled engine is readily to be had when the necessity for this size appears, and it is likely that the air-cooled field will be expanded within a few years. A new Curtiss engine of the V-1400 type has just lately passed its type tests with a power output about equal to the Packard 1A-1500 and on a smaller weight. The numerals used in these designations represent the displacement of the engine; and here is seen a growing tendency to rate engines on a displacement basis.

WIDE VARIETY OF ENGINES NEEDED

The above-mentioned engines form a rather formidable array, but engines of these powers are necessary for the types of aircraft in use. In general, the Navy requires fighting, observation, bombing, scouting, torpedo, and patrol airplanes. Of these, the fighter is a high-speed machine of small radius of action but high rate of climb and high ceiling. The observation plane for gun spotting has medium radius and medium ceiling, and medium rate of climb. The fighters and observation airplanes are in the small-plane class. The bombing, scouting, and torpedo airplanes are grouped in one type of three-purpose machine in the Navy. These are relatively slow-speed, heavy-duty machines of relatively low ceiling and rate of climb but large useful load. The modern patrol airplane takes the form of the large flying boat, usually multi-motored, capable of considerable radius of action over the sea, low rate of climb, and low ceiling but good seaworthiness.

The fighter demands from 300 to 500 hp. and utilizes high crankshaft speeds. These are permissible in airplanes having high

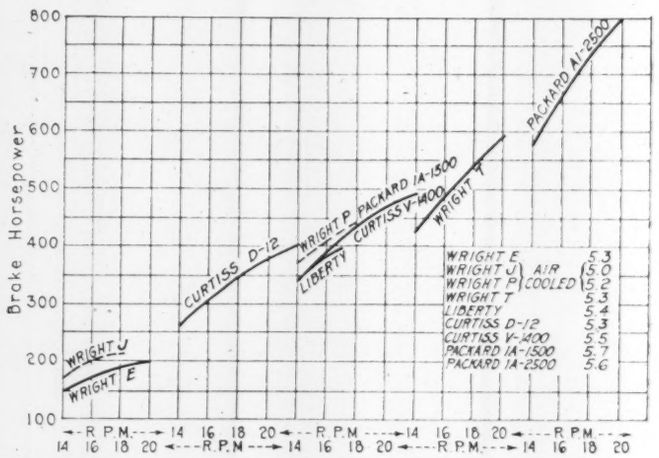

FIG. 2 SERVICE RATINGS OF EXISTING U. S. NAVY AIRCRAFT ENGINES

speeds of advance, but are precluded in slow-speed airplanes on the basis of propeller efficiency. Our observation airplane takes from 200 to 400 hp. and is a medium-duty machine. The three-purpose bomber-scout-torpedo airplane requires from 400 to 800 hp. in a single engine. The patrol airplane requires from 800 to 1200 hp. in at least two engines. The air-cooled engine is used particularly in fighters and observation airplanes because of its lesser vulnerability.

COMPRESSION RATIOS

In comparing engines on a horsepower basis, confusion is likely unless the speed of rotation, the compression ratio employed, and the fuel used are defined. A better measure is the brake mean effective pressure, since this is really a measure of the employment by the mechanism of the cylinder capacity available. Of course, the brake mean effective pressure varies with the crankshaft speed, but to a much less degree than does the horsepower. The influence of the volumetric efficiency on the brake mean effective pressure is, to a degree, a measure of the efficiency of the mixture induction system. Since, however, the brake mean effective pressure is fairly constant over a given range, it gives a good measure of the engine itself.

The specific fuel consumption depends to a degree upon the engine design, and to a still greater degree on the compression ratios used. At the higher compression ratios detonation is encountered, and this occurs in some engines more readily than in others. At present the compression ratios are limited by the character of the fuel available to a normal of about 5.3. Even at this ratio there is some detonation, and in settling on the compression ratio to be realized, a number of items must be balanced. Thus, in an engine having a comparatively high compression ratio there will be obtained in dynamometer tests a higher power and higher fuel economy than can be obtained with a slightly lower ratio. At the same time, however, the detonation resulting from this higher ratio may so affect the engine as to result in its developing lower powers and lower economies after a period of time. For service operation, therefore, a somewhat lower ratio with a correspondingly lower economy and power must be accepted in order that the average economy and power over the life of the engine may be increased.

We are endeavoring to develop a fuel to overcome this limitation. Ethyl fluid has been used successfully. Benzol can be used unless the temperatures encountered are so low as to cause it to crystallize. A blend of benzol, alcohol, and gasoline is fairly satisfactory, but the ideal solution from a service viewpoint is one that eliminates doping or blending. So far, we have not obtained this fuel and are confining ourselves in service to domestic aviation gasoline.

CYLINDER DESIGN

Since aircraft-engine design departs somewhat radically from automobile design, it will perhaps be of interest here to investigate some of the details. Just as the heart of the airplane is its power plant, so is the heart of the power plant its cylinder. The cylinder must measure the mixture to be burned, act as a guide for the

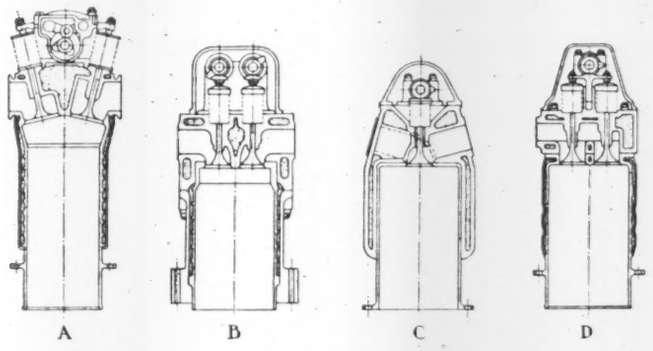

FIG. 3 TYPICAL CYLINDER CONSTRUCTIONS, U. S. NAVY AIRCRAFT ENGINES
A—Liberty cylinder construction. Wet cylinder barrel and welded-on jacket
B—Curtiss D-12 cylinder construction. Wet cylinder barrel and cast-aluminum jacket
C—Wright E and H cylinder construction. Dry cylinder barrel and cast-aluminum jacket
D—Packard 1500 and 2500 cylinder construction. Wet cylinder barrel and welded-on jacket

piston, and transfer a great quantity of heat at a very rapid rate to the cooling system. In general, the power output per unit of volume per cylinder and its efficiency increase with cylinder size. Aircraft-engine cylinders take two general forms, the individual cylinder, as exemplified in the Liberty, and the block construction, as exemplified in the Hispano-Suiza and Curtiss D-12. The weight restrictions in aircraft engines have resulted in the development of four general types of cylinders, shown diagrammatically in Fig. 3. These designs have been evolved not only with the viewpoint of keeping down the weight of the cylinder itself, but also with a view to reducing the overall length of the engine through small cylinder spacing. Since the weight of the contained water also enters into the weight of the engine, this must be kept to the minimum. These requirements have resulted in an early abandonment of cast iron as a material. Since aluminum does not take up the piston wear, all aircraft cylinders are now supplied with steel sleeves.

Referring to Fig. 3-A, the individual Liberty cylinder, the cylinder itself is a steel forging with a welded-on thin steel jacket, and provision for the cam housing at the top. This particular cylinder has the disadvantage of distortion in the portions above the valves and of excessive weight in these non-stressed portions.

The Curtiss D-12 construction which, like the cylinder first shown, is a wet-sleeve type, replaces the steel above the valve seats with an aluminum valve housing and substitutes an aluminum jacket for the thin steel jacket of the Liberty. It will be noted, too, that this cylinder employs two camshafts to actuate its valves. This cylinder has proved very satisfactory except that the end cylinders of the block construction have cracked through the jacket flange, necessitating its being reinforced, as shown in the figure.

The Wright cylinder construction, which is a development of the Hispano-Suiza type, is shown in Fig. 3-C. It will be noted that this design uses the aluminum valve housing and also an aluminum jacket of the dry-sleeve type. This cylinder has been successful but it requires a very nice fit between the sleeve and the aluminum head in order to avoid introducing a serious junction resistance in the path of the heat flow. In the diagram shown the sleeve is closed at the head, but this type has been superseded in later designs in which aluminum-bronze valve seats are inserted into the aluminum valve housing. This cylinder has the disadvantage of considerable distance between centers, especially when multiple blocks are utilized.

The new Packard design, shown in Fig. 3-D, follows the Liberty practice below the valve seat. Above the valve seat, however, it utilizes the aluminum construction. Thus, the cylinder combines all the advantages of the Liberty type with all the advantages of the Hispano-Suiza type, and this development is responsible to a great degree for the weight saving in the new Packard engines. This cylinder utilizes four valves, but only one camshaft is used and the valves are placed in the athwartship line instead of the fore-and-aft line. All of the four types of cylinders are being successfully used in water-cooled engines.

In the air-cooled engine, early cylinders were of aluminum with a forged-steel sleeve to take the piston wear. The aluminum flange for bolting the cylinder to the crankcase did not stand up very well, and as a result the modern cylinders employ an aluminum head carried down fairly close to the flange, which is now a part of the steel sleeve as in Fig. 4. The detailed design of the air-cooled cylinder is still under investigation. To date it has not been possible to carry the high brake mean effective pressures in air-cooled cylinders that are carried in the water-cooled. However, these are well on the way to attainment. The primary requisite is a close thermal contact between the steel sleeve and the aluminum head, careful valve-seat cooling, proper spark-plug location, proper relation between the inlet- and exhaust-port locations, and proper combustion-chamber form. One cylinder design employed the use of a relatively short aluminum head with steel fins on the exposed portion of the liner. In one comparison with a similar aluminum-fin type, almost exactly the same performances obtained. Since, however, the steel-fin construction is a more difficult shop problem than the aluminum-fin type, the latter has been adopted in the Navy. The aluminum heads can be cast without additional machine work. The machining of the steel fins involves greater care because of the machining and the spoilage which results. There is still much room for improvement in air-cooled cylinders, and experimental work is being continued along this line.

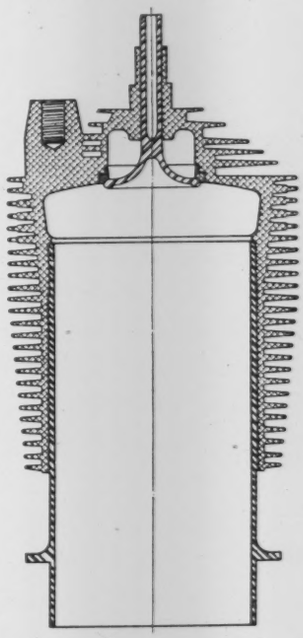

FIG. 4 CYLINDER CONSTRUCTION OF AIR-COOLED AIRCRAFT ENGINE
(Steel cylinder barrel with cast-aluminum head.)

VALVES AND METHODS OF COOLING THEM

The design of the inlet and exhaust valves is very closely associated with that of the cylinder. Where aluminum heads are used, the valve seat is ordinarily of aluminum bronze, on the basis that the coefficient of expansion of this material is very similar to that of aluminum itself. However, these valve seats must be carefully inserted and they are generally rolled in with a view to improving the thermal contact. Special materials are required to withstand the high temperatures encountered in exhaust valves for aircraft engines. At present we prefer silchrome steel for valve material. The design of the valve is very important, and relatively small changes in proportions produce widely divergent results in performance. There are two general types of valves—the mushroom and the tulip type—and of these two the latter is in more common use.

In the attempt to eliminate the strict material requirements in exhaust valves, designs have been developed for cooling these valves. For this purpose mercury was used in early design in which the hollow valve stem containing the mercury was sealed

with a plug at the top. The purpose of the mercury was, of course, to transfer the heat from the valve body to the valve stem, with a view to carrying it away through the guide into the cooling water. All these valves containing liquid are subjected to high pressures, which makes the valve stem sealing difficult and sometimes results in bulging the valve stem.

This led to the employment of certain salts for cooling purposes. The Engineering Division of the United States Army Air Service has sponsored this development with surprising success. The salt used, a eutectic mixture of sodium and potassium nitrates, produced little or no pressure in the valve but material assistance in the transfer of the heat. In a late design such a valve plated

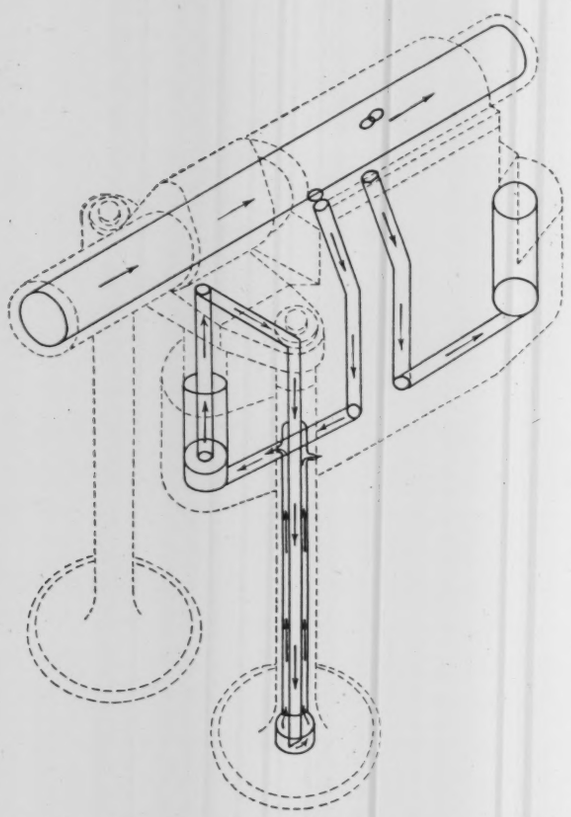

FIG. 5 PACKARD 1 A-1500 OIL-COOLED EXHAUST VALVE

with chromium has proved very successful. With these salt-cooled valves, the heat transfer through the valve-stem guides is large and the corresponding valve-stem guide wear may bring about the necessity for hardened guide material. The use of this material complicates repairs in the field, however, and to date salt-cooled valves have not been widely adopted.

Another attempt at the solution through cooling is that in the new Packard engines in which the lubricating oil is circulated through the valve by a small mechanical device, Fig. 5. So far, this method has given good service and is not subject to the apparently obvious objection that the cooling system becomes plugged up with foreign matter. When the cooled valves are used in an engine alongside the uncooled valves, a marked difference is apparent to the eye. The uncooled valves run red hot, while the cooled valves are dead black. Cooled valves are naturally of longer life than hot valves.

PISTONS

Many experiments have been conducted with a view to improving piston design. Manifestly the piston must transfer large quantities of heat to the cylinder, and it must provide an adequate bearing surface to take up the connecting-rod thrust. To meet these conditions, pistons must be thick enough to avoid reaching too high a temperature. Most of the heat is transferred through rings to the cylinder wall, and for this purpose we use the ordinary snap ring peened on the inside.

Early pistons were very long, but later pistons have been much reduced in length. The slipper-type piston, in which much of the material at the front and back sides is entirely cut away, has been used, but these have produced excessive liner wear in some cases. Under the auspices of the National Advisory Committee for Aeronautics, the Bureau of Standards is conducting some interesting tests on piston friction. These tests are not yet completed but they are bringing out some interesting factors in piston design. Piston friction, of course, forms a large proportion of the internal friction of the engine, and these tests will point out the factors involved.

Aircraft engines are not subjected to the same piston conditions as are automobile engines. The matter of oil pumping is not so important because the engines run hotter and carbon is not such a problem. Again, lubricating oil in aircraft engines is changed more frequently and the problem of crankcase dilution is not so pressing. The noise of piston slap is unimportant. The important problem is to get away with the minimum weight of material.

Along these lines, it is important to note that all aircraft engines use an aluminum alloy for piston material. Tests recently conducted with pistons, with 87 per cent magnesium and 13 per cent copper, indicate that these are even better than the aluminum pistons when properly designed, because the magnesium seems to stand up a little better under the detonating conditions at which aircraft engines are frequently run. Other tests indicate that the use of magnesium-copper alloy for piston-pin bushings is satisfactory. Of course, one bushing itself does not constitute a great part of the engine weight, but it is interesting to know that the reduction in weight of the Liberty engine through the use of these bushings is about 3.8 lb.; there being twelve cylinders in this engine. Improvements in piston design, then, are based around the use of a more suitable material with a better design.

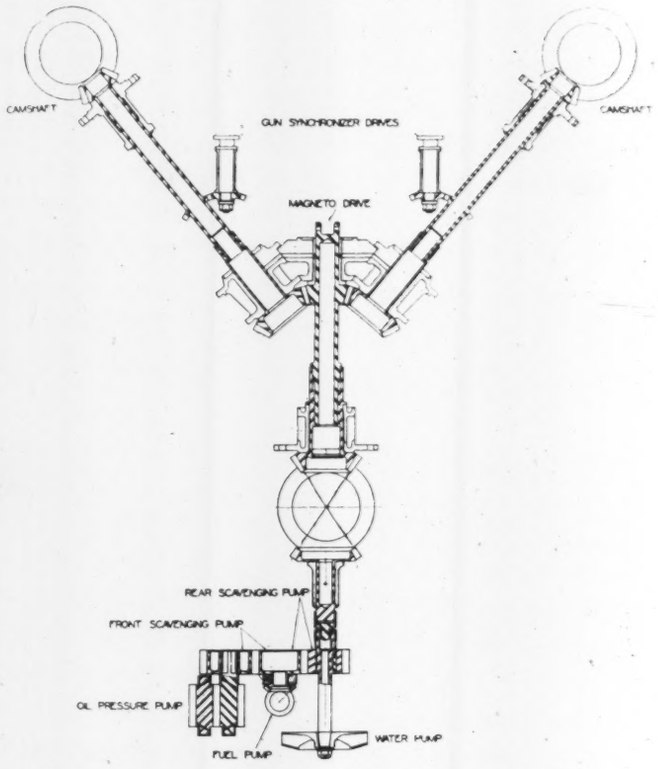

FIG. 6 DIAGRAM OF TIMING GEAR AND ACCESSORY DRIVE LAYOUT, PACKARD MODELS 1500 AND 2500

CONNECTING RODS

Two types of connecting rods are in general use in aircraft engines—the forked type and the articulated type. Of these, the former type is in more general use. The difficulty with the forked type is that the impulse on the blade rod comes too near the face of the bearing, thus introducing problems. These rods are also difficult to service. Some difficulties have been had in the design of the articulated rods, but all these difficulties are now apparently overcome except the difficulty of eccentricity, which is inherent.

All rods are made of special steel alloy designed for the maximum strength on a minimum weight.

Bearings

Considerable improvement has been brought about in the design of aircraft-engine bearings. Early practice for main bearings incorporated a bronze shell with a babbitt surface. A late development, which has produced a very great improvement in the Liberty engine particularly, is that of the steel-backed main bearings. The greater stiffness of this backing material not only serves to avoid high spots under running conditions, but also serves to stiffen up the crankcase to such a degree that main-bearing life is very greatly improved.

For connecting-rod bearings where the forked-type rod is used, the general practice is to use a special bronze, such as "Kelly metal." This has proved very satisfactory. When an attempt is made to use the steel-backed bearings in the connecting rods of the forked type, it becomes necessary to use either babbitt or bronze on the back of the bearing. It has been found possible to cast the bronze on the steel, but to date these bearings have not been entirely satisfactory because of the peculiar conditions under which the blade rod operates. Steel-backed bearings can be readily utilized in the articulated-type rod, and the newer engines incorporate articulated rods for this very reason. In the air-cooled engine the steel-backed bearing for the big end of the connecting rod has given immediate success.

In the early designs, aircraft-engine crankshafts were made much too small. They were capable of withstanding the explosion loads put upon them, but the severe torsional vibrations set up in an engine of this kind and the whip which the shaft encountered, forced us to considerably larger crankshafts. Another consideration is that it is extremely difficult to get, in small quantities, the high quality of material used for small-diameter crankshafts. In the interest of crankshaft stiffness and commercial production it was necessary to go to somewhat larger sizes, and the modern aircraft-engine crankshaft is brutally heavy.

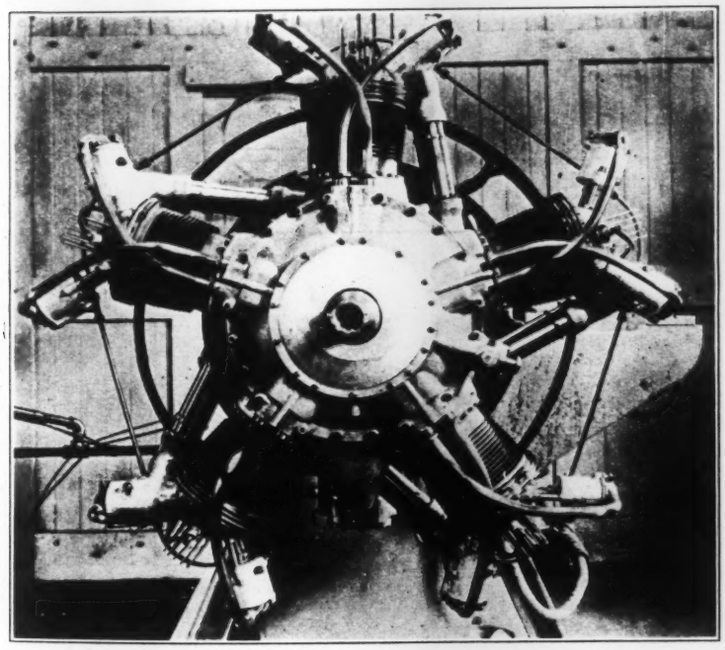

Fig. 7 Air-Cooled Aircraft Engine Equipped with "Oleo" Hydraulic Valve Gear

Crankcases

To obtain stiffness in crankcases without excessive weight, many of the later engines incorporate the arrangement wherein the parting face of the upper and lower halves is brought considerably below the shaft center line. The upper half, then, carries the main bearings and the lower half becomes simply a light pan of cast aluminum. All crankcases are of aluminum alloy.

Auxiliary Drives

Aircraft engines usually incorporate five accessory drives, the camshaft drive, two magneto drives, a water pump drive, and an oil-pump drive. In the case of the new Packard engine, Fig. 6, these drives have been reduced to three, the camshaft drive, a drive for the double magneto, and the pump drive, of which one pump drives another. In the case of the camshaft drives it became necessary to utilize much larger and heavier gears than was at first anticipated. This is due to the fact that these drives, for the sake of accessibility, are mounted on the anti-propeller end and the torsional vibrations subject them to shock loads which they would not otherwise suffer.

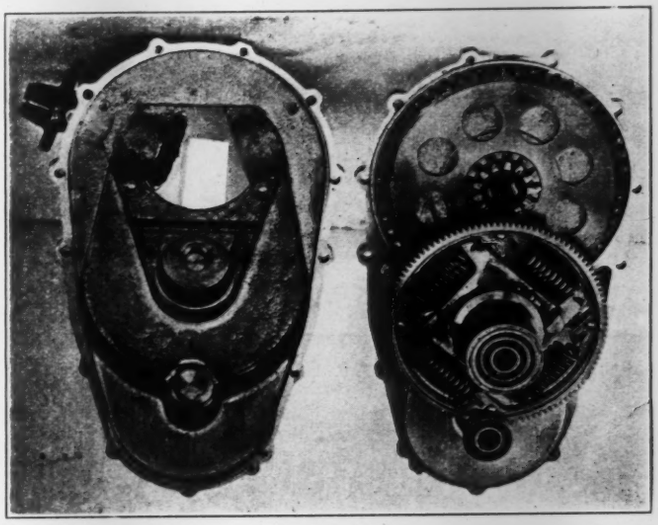

Fig. 8 Flywheel Starter of Aircraft Engines
(Casing removed showing constant-torque slipping clutch, calibrated springs, and tangent screw adjustment.)

Overhead camshafts and overhead valves are used almost exclusively in aircraft engines because of considerations of weight and combustion-chamber form. In the static radials, push rods are used, and a number of difficulties have been encountered which are, of course, not present in the water-cooled type. Considerable trouble was encountered in the Liberty with broken camshaft housings because these were too light to tie the cylinders together properly. All American engines subsequent to the Liberty utilize a bigger and more rigid camshaft housing with a light removable cover. In the effort to eliminate the push-rod difficulties of the radial, the Kinney Manufacturing Company, of Boston, has developed an "oleo" valve gear which is of the hydraulic type and shown in Fig. 7 applied to an engine. It has been very successful and has possibilities of employment in automobile as well as aircraft engines.

Engine Starters

Ordinary hand starters are satisfactory for small engines, but for the larger engines a power starter seems necessary, particularly in cold weather. Considerable development has been made with the electric starter, but the large current requirement of such a starter is a handicap. A new Navy development, known as the "inertia" starter, Fig. 8, was developed by the Aeromarine Plane and Motor Company, Keyport, N. J. In it, by turning a hand crank which drives a flywheel through gearing, high speed is imparted to the flywheel and energy is stored up in it. By a constant-torque slipping clutch this energy can be transferred through gearing to the crankshaft, rotating the shaft at quite high speed. This starter for aircraft engines has been uniformly successful and is considered to be one of the outstanding developments of the year. It has been very satisfactory even in the large Packard 2500 engine, giving starts with a dead-cold engine at will. With it the ordinary booster magneto generally used will be eliminated and the engine can be started with the main magnetos. It is of

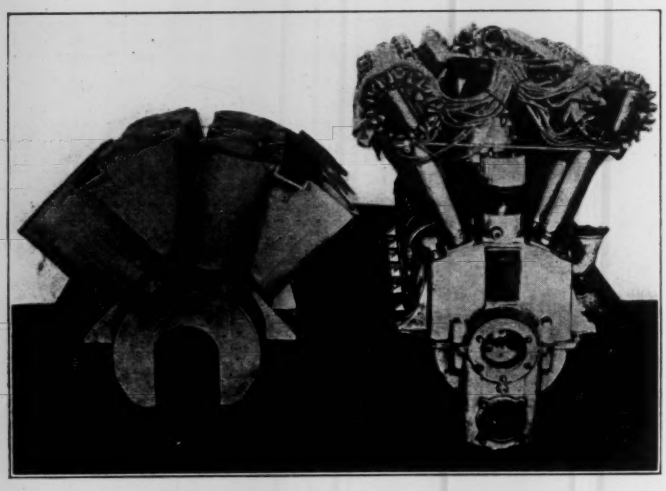

FIG. 9 COMPARISON OF TWO 12-CYLINDER MOTOR ENDS

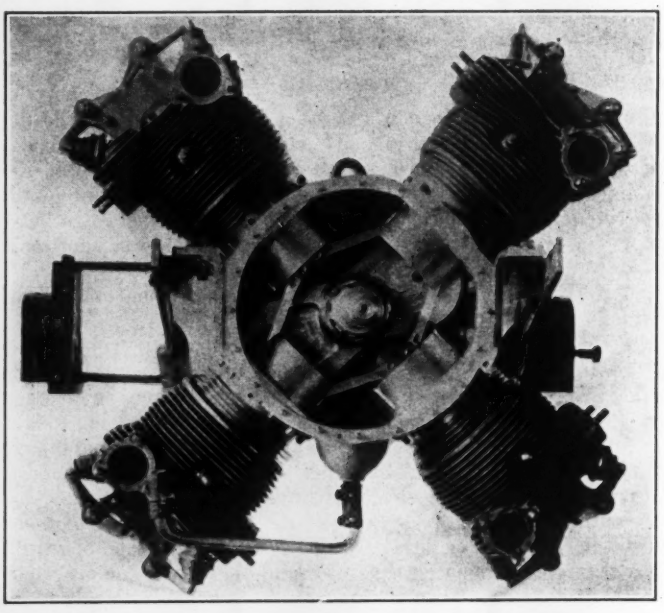

FIG. 12 CRANKLESS "CAM" AIRCRAFT ENGINE

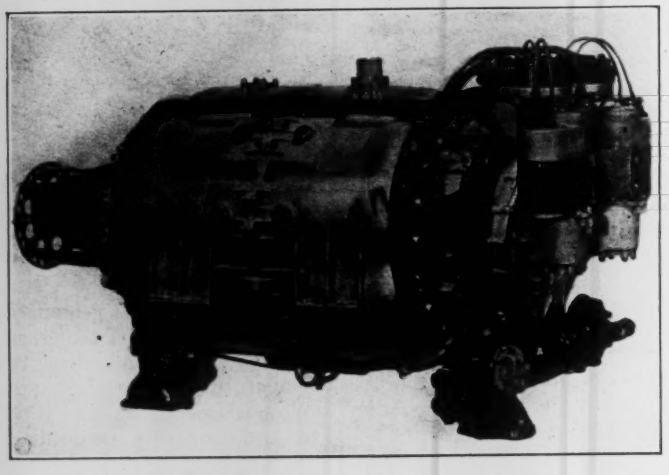

FIG. 11 ALMEN "BARREL-TYPE" AIRCRAFT ENGINE

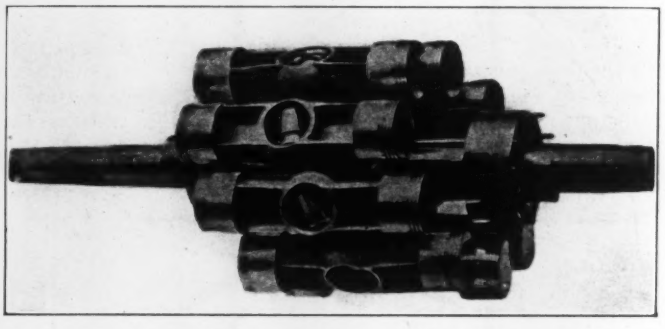

FIG. 11a PISTON ELEMENT OF THE "BARREL-TYPE" ENGINE

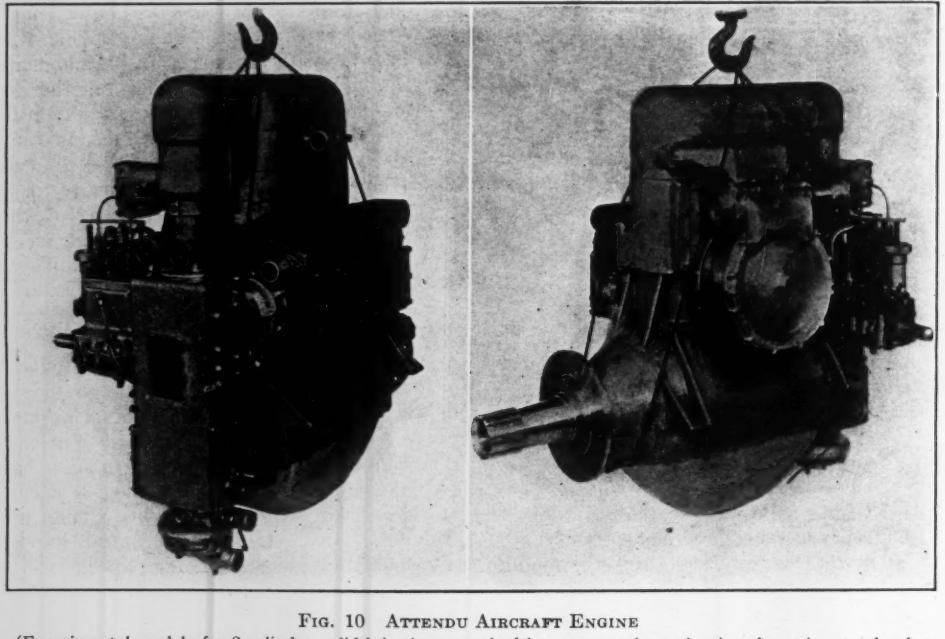

FIG. 10 ATTENDU AIRCRAFT ENGINE
(Experimental model of a 2-cylinder, solid-injection, auto-ignition, two-stroke-cycle aircraft engine to develop 125 hp. at 1800 r.p.m. Weight, 3.5 lb. per hp.)

comparatively light weight, and under favorable conditions there is enough energy stored in the flywheel to permit a number of starts on one crank. This starter has been adopted for general use in the Navy.

Future Developments

So far, aircraft-engine development and progress to date have been outlined. It now becomes interesting to estimate the future of developments. In general, the rise of the air-cooled engine is pushing the water-cooled engine to greater achievements. In order to overcome the limiting propeller speed, we are resorting to gearing in water-cooled engines. Gear ratios of 5 to 3 and 2 to 1 are being used. These gears permit higher crankshaft speeds for a fixed propeller speed and a corresponding increase in power output which may, in the near future, bring the power-plant weight of the water-cooled engine to a point comparable with that of the air-cooled engine in spite of the additional weight of the cooling system. In the attempt to run existing aircraft engines of higher speed, certain modifications have been made in the induction system to keep up the volumetric efficiency and the brake mean effective pressure at high speeds. Beyond the point where this can be done mechanically, resort may be had to a rotary induction system. The tendency in water-cooled engine design is, then, to higher crankshaft speeds with reduction gears. The gears themselves have presented a number of problems because of the torsional vibrations and whip of the crankshaft, but these difficulties have been met through the employment of a spring coupling in the gear system, and satisfactory gears are now available.

Air-Cooled Liberty Engine

A late development by the Air Service Engineering Division at McCook Field is an air-cooled Liberty engine. This brings to mind the fact that we are prone to consider a water-cooled engine as necessarily a V-type or in-line engine, and an air-cooled engine as necessarily a single-crank or two-crank static radial. As a matter of fact, either of these types can be used for either air or water-cooled engines and they are now being used. As engines increase in size, resort can be had in the water-cooled field to the use of the W or broad-arrow type in which there are three lines of cylinders on the top of the engine. In the X-type there are four rows of cylinders, two above and two below the center line of the crankshaft. The general tendency, then, is to shorter engines utilizing more lines of cylinders, and it would seem that both the air- and water-cooled engines must some day approach the same general form.

The air-cooled Liberty referred to above has been successful, at least in so far as its cooling is concerned. In it, more energy is required to blow the air between the in-line cylinders than is the case in the single-crank radials, and the cylinder spacing of the air-cooled in-line engine must of necessity be greater than that of the water-cooled in-line engine. The air-cooled in-line engine has great possibilities and will enter into future developments.

Other Cylinder Arrangements

Under Navy supervision the Aeromarine Plane and Motor Company has developed a new engine in which four rows of cylinders are mounted on the top of the engine and are staggered so as to reduce the frontal area, as in Fig. 9. This is a 12-cylinder engine so that only three cylinders constitute a row, making the engine much shorter. Obviously, this engine can be expanded to 24 cylinders on the same cylinder dimensions, giving a 5000-cu-in. engine capable of developing 1500 hp. on overall dimensions only slightly greater than those of the present V-types. If water-cooled engines of this power are required, this type has immediate possibilities. Sooner or later, we will probably have water-cooled engines of 1000, 1200, or 1500 hp., and air-cooled engines of 350 and 650 hp. It is interesting to note, in connection with power plants, that necessity for the power plant often arises after the power plant has been developed. Those responsible for power-plant development must foresee this necessity and provide power plants. In the present fluid state of aircraft development, power-plant people are rather hard pressed to determine just what line they should pursue.

Development of New Type

The development of new types ordinarily covers about two years. There are three stages of development, the experimental, the flight testing, and the production stage. We generally contract for an experimental model in numbers of three to five. These engines are thoroughly tested on the block with a view to getting all the "bugs" out of them.

A limited production order is generally placed for about a dozen additional engines, which are immediately put in aircraft for flight testing. Almost invariably certain defects become apparent that were not noticed on the block, and the purpose of the flight testing is to eliminate these defects before going into production. After the engines are in production, they are carefully watched by the power-plant people. Trouble reports are required from the operating personnel, and it is natural that engines which have met the requirements in flight testing develop defects under operating conditions where the average skill of the personnel is not so high and the operating conditions are more difficult. While the engine is in production, then it undergoes numerous refinements and changes from order to order. Unfortunately, about the time an engine has reached its maximum development the advance in aeronautics puts it out of the picture for one reason or another, so that the production life of a given model has been limited. As time passes, however, we become more and more standardized and it seems likely that the development of new types will not take place so rapidly as it has in the past.

Heavy-Oil Engines

The engines discussed so far have been gasoline driven and have used high-test gasoline at that. The problem of using the heavier oils is a most difficult one and so far has not been entirely solved. Just lately, however, a most interesting development has been brought about by the Navy. This is known as the Attendu engine, Fig. 10, and is manufactured by the Eastern Engineering Corporation, Limited, of Montreal. The experimental model is a two-cylinder solid-injection, auto-ignition, two-stroke-cycle aircraft engine designed to develop 125 hp. at 1800 r.p.m. The engine as delivered weighs $3^1/_2$ lb. per hp. It is flexible from idling speed to full throttle. It starts quite as readily as an aircraft engine. It runs with a clear exhaust, burning a very heavy oil (19 deg. Baumé). The engine is now on test at the Aeronautical Engine Laboratory at the Naval Aircraft Factory at Philadelphia and has given surprising results. Even though this engine has not yet met all its requirements, one of which is a fifty-hour full-throttle test, it has already demonstrated some surprising characteristics. Its first application would be in rigid airships where, by reducing the fire hazard due to gasoline vapors, the hazard of operating such ships can be greatly reduced and hydrogen as a lifting gas may be used for some purposes.

"Barrel" and "Cam" Engines

Two interesting developments are being carried on by the Engineering Division of the Army Air Service at McCook Field. One of those is the Almen "barrel" type, Fig. 11, engine which utilizes the swash-plate principle. Its troubles are primarily bearing problems, but these seem on the way to solution and this type has interesting possibilities in weight saving and space saving. Another Air Service development is the so-called "cam" engine, in which a cam replaces the conventional crankshaft, Fig. 12. The problem here is in the design of the cam and the material of which it must be made, but this engine also has interesting possibilities for the future. Experience indicates, however, that most unconventional engines have some complication which limits their application, and the developments of the future will probably follow two lines: first, the continued refinement and improvement of the conventional types and second, experimentation with unconventional types.

On this basis, we need not feel that we have approached the limits in aircraft engines. Any one, in this stage of development, who for a moment is inclined to feel satisfied with what we have, is just about ready for a severe jolt. In this game, we try to keep our heads in the air and our feet on the ground, but since we are standing on quicksand, our position is somewhat unstable.

Aeronautical Engineering at McCook Field

Developments in Aircraft Engines—Air-Cooled Aircraft Engines—Generator Problems Due to Crankshaft Vibration—Fatigue of Welds—Earth-Induction Compass—Magnesium and Its Alloys

EN ROUTE to the Spring Meeting of the A.S.M.E. at Milwaukee last May, members of the Society were guests of the U. S. War Department Air Service at McCook Field, Dayton, Ohio, on May 16. The visit afforded an opportunity for an inspection of the workshops, laboratories, and flying fields; and the character of the engineering and research work done under the direction of the Engineering Division of the Air Service as well as some important developments in aeronautical engineering were presented in a series of short addresses by engineers attached to the Division. Six of these addresses follow.

Achievements of American Aircraft-Engine Industry

By E. T. JONES,[1] DAYTON, OHIO

CONDITIONS immediately following the World War necessitated a complete reorganization of the American aircraft-engine industry. Many companies which had expanded greatly during the war found it necessary to reorganize on a much more restricted basis, while some companies undertook a considerable expansion based on over-optimistic estimates of the immediate future of commercial aeronautics. It was natural that during this period very little real progress was made in the development of aircraft power plants, but as the industry gradually stabilized into a group of fairly permanent organizations, progress became much more rapid and during recent years very substantial gains have been made. The outstanding features of this development have been a great reduction in engine weight accompanied by an increase in reliability; the development of larger engines; the working out of the fundamentals essential to the production of satisfactory air-cooled engines of any desired power output; the development of highly satisfactory reduction gearing; and the successful solution of the supercharging problem.

The greatest success in the development of large engines has been attained by following closely the design features worked out in the smaller sizes.

The Engineering Division of the Air Service and other similar agencies are continuously flooded with inventions aimed at radical improvements in engine performance and numerous patents have been issued since the war based on schemes which the inventor confidently believes will revolutionize the entire development of aircraft power plants. In spite of this flood of inventive genius it may be safely stated that none of these patents or inventions may be considered, to an appreciable extent, responsible for the remarkable improvements which have been made, nor is there a single case in which equally satisfactory results could not be obtained by eliminating the few patents or inventions which have been applied in aircraft-engine design. The most successful aircraft engines at the present time may be considered as careful refinements of the types available at the end of the war. These refinements consist mainly in the development of designs of greater compactness, which naturally leads to lighter weight and greater rigidity; the employment of higher crankshaft speeds; improvements in valve-operating mechanism, valve cooling, and valve materials; improvements in bearing design; and improvements in volumetric efficiency.

Of course, development along the fixed lines mentioned cannot proceed indefinitely and unless new principles are applied or new materials developed, progress must eventually cease. However, there is no reason to believe that we are approaching such a condition at present as there still appears to be prospects of very material gains along the conventional lines. Just at present the most promising line of development appears to lie in the direction of higher crankshaft speeds. The racing automobile has indicated great possibilities in this direction, but it must be remembered that crankshaft speed is a function of engine size and that the same type of design which will permit satisfactory operation at 5000 r.p.m. in the 122-cu. in. class cannot be expected to give the same results when applied to engines of from 1500- to 3000-cu. in. displacement. It must also be borne in mind that unlike the racing automobile the aircraft engine is not limited as to displacement, and increasing crankshaft speed is useful only when accompanied by a reduction in specific weight of the power plant.

New Engine Types

While remarkable results are being achieved by adhering closely to conventional types of construction, the Engineering Division has not entirely disregarded the possibilities of other mechanisms being developed which may be superior to the conventional crankshaft, connecting-rod, and poppet-valve combination. Two novel types are being experimented with, both of which appear to offer possibilities of eventually surpassing the conventional type. In the water-cooled engine class the Almen barrel-type engine[1] has been developed to a point where it may be confidently stated that it can at least equal the conventional type as regards weight, bulk, and reliability, and due to the inherently compact arrangement and general rigidity of the design there appear to be excellent possibilities that this type can be developed into a lighter and more compact engine than any now available. This engine also possesses the advantage of utilizing considerably fewer parts and being much easier to assemble and disassemble than the conventional engine. In the air-cooled field the "cam" engine[1] is being developed. This engine also eliminates the conventional connecting-rod and crankshaft arrangement. At present the chief advantages of this type of engine appear to lie in the possibilities of reduced complication and cost rather than in greatly reducing the weight. This arrangement gives perfect running balance in a four-cylinder engine and thus permits smooth operation at higher outputs than have been possible with four cylinders. This of course tends to lower the cost of the engine by reducing the number of parts and it is also believed that the cam and roller mechanism can be produced at a lower cost than the crankshaft and connecting-rod arrangement which it replaces. The fact that the piston makes four strokes for each revolution of the main shaft gives this engine the advantage of a 2 to 1 reduction gear without any increase in weight or complication. This engine is still in the very early stage of development but has progressed far enough to demonstrate that the mechanism can be successfully operated; however, the development has not yet progressed far enough to indicate definitely how this type will compare with the conventional engine as regards weight per horsepower. Another development which may be briefly mentioned is the successful operation of inverted engines. While this idea is by no means new, it is only during the past year that engines of this type have been successfully used in service airplanes. The chief advantages of this arrangement appear to be improvement in visibility forward, easier disposition of the exhaust gases, higher location of the propeller center, and better accessibility to the carburetor, spark plugs, valves, etc., from the ground.

Reduction Gearing

The introduction of reduction gears in aircraft power plants has progressed rapidly during the past few years. This development has been rendered essential by the increased size of the engines and the higher crankshaft speeds employed. The use of gearing has also been made more attractive by reduction in its weight, largely brought about by recent improvements in methods of gear manufacture. At the close of the war the introduction of reduction gearing necessitated a considerable increase in the weight of the power plant, while at present we have highly reliable gears which add somewhat less than 0.2 lb. per hp. In this field the aircraft

[1] Engineering Division, Air Service, McCook Field.

[1] An illustration of this engine will be found on p. 788.—Editor.

industry has merely taken advantage of improvements which the gear-cutting industry at large have recently made in the art of gear manufacture. Analysis of several typical aircraft reduction gears shows tooth speeds in the vicinity of 3500 ft. per min. accompanied by tangential loads of 2000 lb. per in. of face width. It is only recently that gears combining hardness, strength, and accuracy sufficient to permit operation under these severe conditions have been available. Most successful aircraft reduction gears have been protected from the ill effects of engine-torque variation by the introduction of a shock-absorbing coupling between the crankshaft and the driving pinion.

SUPERCHARGERS

The development of superchargers has progressed continuously since the war and here again the work has consisted mainly of painstaking refinement of the general type of design originally conceived rather than by introduction of new and novel types of mechanism. The result of this work is that we now have turbine superchargers in service which are quite as reliable as other parts of the aircraft power plant. The gear-driven supercharger development was started somewhat later than that of the turbine type and has not yet resulted in a design considered sufficiently reliable for service use. However, considerable success has been attained and it is confidently expected that superchargers of this type will be available for service use within the next twelve months. The chief problem in connection with this development has been providing a satisfactory light-weight driving train which would withstand the violent accelerations to which the aircraft engine is subject. This appears to have been solved by carefully reducing the inertia of the high-speed impeller to a minimum and by utilizing a flexible spring coupling between the crankshaft and the supercharging gearing. It has been found that the use of superchargers will increase the ceiling of any type of airplane by approximately 80 per cent and will greatly improve the speed and maneuverability at altitude.

From this brief summary of the present status of aircraft-power-plant development it is evident that Americans may well look with pride to the achievements of our engineers and manufacturers and may be sure that in practically every class of power-plant equipment American products are equal, if not superior, to the best products of any other nation. We must not, however, assume that we have reached a point where we can relax our efforts toward further improvements, as it is only by exerting ourselves to the utmost that we may expect to maintain our present position.

ENGINEERING DIVISION OF THE AIR SERVICE

The functions of the Engineering Division in connection with the developments discussed above are confined to basic research, development testing, financing of experimental developments, and supervision of experimental contracts. Every effort is made to avoid the complete development of service equipment by the Engineering Division personnel working in the Engineering Division shops as it is quite as important to develop a satisfactory source of supply as it is to develop the equipment itself. Therefore, in speaking of work being done by the Engineering Division we necessarily include work being done by all contractors, and the major portion of the credit for the results which have been accomplished must necessarily go to them as they carried out the majority of the work. The Engineering Division's chief task consists in seeing that the proper lines of development are followed by giving financial encouragement to those developments which appear desirable and by withholding such encouragement from developments which do not appear to merit the financial assistance of the Government. The actual development is thus largely in the hands of the industry, while the determination as to what lines it shall follow is largely in the hands of the Engineering Division. With this in mind we may speak of our developments as including all development done by the contractors under Engineering Division supervision without appearing to take undue credit for work which was actually accomplished by the industry.

Among the outstanding achievements in which McCook Field coöperating with the contractors has taken the lead, may be mentioned the following: the first American engine in the 700-hp. class; the largest American water-cooled engine in use today; the lightest American water-cooled engine; the highest-speed American water-cooled engine to successfully pass an endurance test; the first successful application of the inverted engine for service airplanes; the first American-built nine-cylinder radial engines in both the 150- and 400-hp. classes; the only successful modern air-cooled V-engine of an output suitable for service airplanes; the only air-cooled cylinders of over 30 hp. which have proven satisfactory from the point of view of reliability, output per unit weight, cooling, and fuel consumption; the first successful application of the rotary induction system to American radial engines; the first air-cooled engine to be equipped with a completely inclosed and thoroughly lubricated valve gear; the largest air-cooled engine in the world; the only successful supercharger to go into service use; all work on the application of gear-driven centrifugal superchargers to American aircraft engines; and the development of a complete line of accessories necessary for satisfactory operation of the supercharged airplane.

The Engineering Division takes considerable pride in this record and hopes to continue in the position of pioneers in engine development in the future. This position is of course not entirely without its drawbacks since pioneering always involves considerable risk and a certain percentage of failures. However, for this very reason it is probably necessary that the pioneering be financed by the Government. If the Engineering Division were to take the position of refusing to participate in a development until private industry or other government agencies had clearly demonstrated its feasibility, development would be materially retarded.

Air-Cooled Engine Development

BY S. D. HERON,[1] DAYTON, OHIO

The development of air-cooled aircraft engines has been in progress by the Engineering Division of the Air Service for the last five years. The first three years of this period was almost entirely devoted to cylinder investigation by means of single-cylinder engines. By the middle of 1922 cooling had ceased to be anything but a minor problem and since then the major aim of the development program as regards cylinders has been to produce a construction that is mechanically durable and that does not deteriorate due to warping, growth, and thermal stresses when subjected to protracted service at full power. Further, it is a fundamental requirement that the construction shall be a good production proposition in the sense that each operation shall be readily open to inspection during manufacture. Any such uncertainty as cast-in valve-seat inserts which can only be definitely inspected for soundness by cutting up the casting containing them, has perforce to be rigidly excluded. Costly and difficult construction is to be avoided as, while efficiency is the aim rather than first cost, nevertheless cost is essentially man-hours and difficult construction restricts manufacturing facilities, both of which are at a premium in time of military stress. Attention has in particular to be directed to the foundry aspect of production, as difficult castings involving superimposed masses of cores seem to be almost unproducible in time of war. The result of the Division's efforts as regards cylinder construction has been to produce a design which, although far from finality, fulfills all the above requirements.

The activities of the Engineering Division in multi-cylinder-engine development have been centered in the engine of 400 hp. and greater. Working in conjunction with the aircraft-engine industry two successful 400-hp. types have been produced. The first of these is the Curtiss R-1454 9-cylinder radial engine shown in Fig. 1. This engine develops 400 hp. at 1650 r.p.m. with much in reserve, and weighs 740 lb., complete, with built-in hand starter and supercharger. This engine has undoubtedly set a new standard in smoothness of running for large single-row radial engines. This is almost entirely chargeable to the supercharged induction system. This engine, which reflects much of the skill of its constructors, has partly finished its development tests and is now being reworked to take the type-M cylinder shown in Fig. 2. The cylinder shown in Fig. 1, while extremely durable, is very crude as regards head resistance and valve gear and was used purely to develop the rest of the engine while a more suitable type was designed and tested.

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The air-cooled V-engine was virtually dead from 1917 onward until revived by the Engineering Division. It was considered that the single-row radial engine was not suitable for powers in excess of 500 hp. on account of excessive head resistance and mechanical difficulties with valve operation, connecting rods, and pistons. While the radial is undoubtedly the lightest possible type of air-cooled engine and is extremely easy to maintain and overhaul, nevertheless in sizes beyond 500 hp. the above-mentioned difficulties are so accentuated as to render the type inferior on all counts to some form of line engine, either V or X.

This Division has reason to consider that the double-row radial with staggered cylinders is inferior to the single-row radial, V, or X types on account of cooling trouble with the rear row of cylinders, and further due to the difficulty of securing a durable and efficient type of valve-operating gear. The result of this attitude was the production of the air-cooled Liberty-12. This engine consists of standard Liberty 12-cylinder water-cooled engine fitted with new cylinders, pistons, and valve operating gear. It is of $4^5/_8$ in. bore by 7 in. stroke (the standard water-cooled type being 5×7). It develops 420 hp. at 1900 r.p.m. and has passed a 50-hr. test, developing 380 hp. at 1880 r.p.m. It weighs 950 lb. which is 65 lb. in excess of the dry weight of the standard water-cooled type; the net installed weight, however, is 200 to 230 lb. less for the air-cooled type. The engine will cool continuously at full throttle while stationary on the ground with the blast supplied by its propeller of normal flight type.

The fact that it is easy to cool the air-cooled Liberty embodying a 45-deg. included angle between cylinder banks, which is about as difficult a type of line engine as can be imagined from the cooling standpoint, at once shows that the 1000-hp. air-cooled engine is in no sense a radical step. If an X type with a 90-deg. angle between the banks of cylinders and 24 cylinders be adopted, it is possible to obtain 1000 hp. or more without the required power per cylinder exceeding that obtained in the Curtiss R-1454 and without any anticipation of cooling trouble. Design and construction of such an engine are already under way.

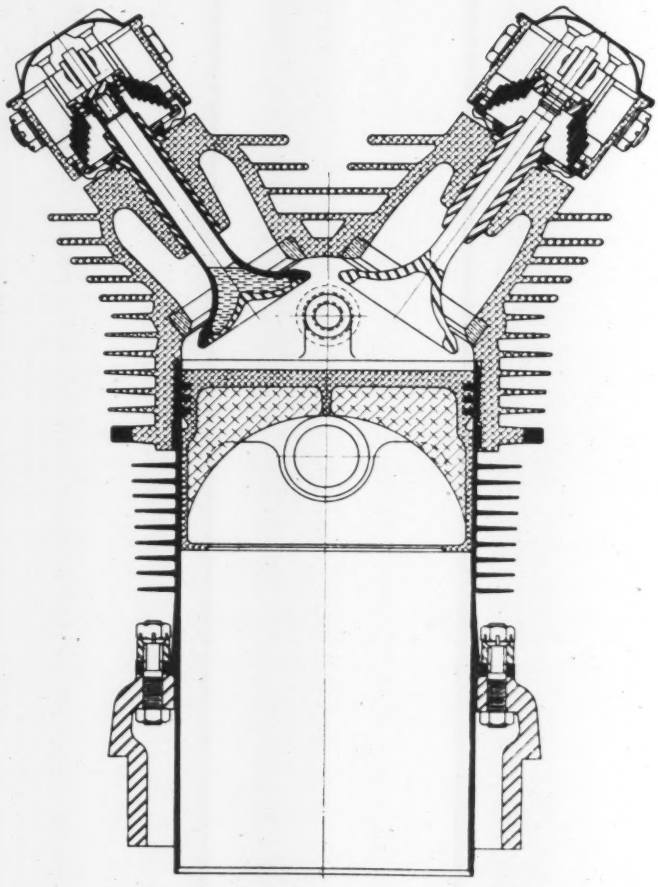

FIG. 2 TYPE-M AIR-COOLED CYLINDER

CYLINDER CONSTRUCTION

The type of cylinder construction developed by the Engineering Division is shown in Figs. 2 and 3. The head is a heat-treated aluminum-alloy casting screwed and shrunk on to a steel barrel having integral steel fins machined from the solid; the valve-seat inserts are of aluminum bronze and are shrunk into the head. A steel clamp ring is shrunk over the lower end of the head so that the latter is nipped between two steel surfaces, preventing stretching and movement of the head on the barrel and eliminating possibility of leakage there. The valves, which are both of glass-hard steel, operate in glass-hard tungsten-steel guide bushings, thus eliminating all wear of these parts. The exhaust valves are internally cooled by means of a mixture of fused sodium and potassium nitrates contained in the hollow stem. The outstanding feature of the type-M cylinder shown in Fig. 2 is the reduction in head resistances compared to the type-J cylinder originally fitted to the Curtiss R-1454 engine. The arrangement of ports and the enclosure of the valve gear result in producing about the minimum possible disturbance of the air passing over the cylinder head. The valve gear is of interest as it is entirely enclosed and lubricated by force-feed oil and is of the compensated constant-clearance type. The compensation is introduced primarily to remove the heavy bending moment to which the cylinder head and barrel are subjected with an uncompensated gear such as used on the type-J cylinder. The elimination of valve-gear shock and timing variation resulting from the compensated gear are of secondary importance compared to the reduction in bending moment on the cylinder. This cylinder was designed in 1923 and considerable development has already been carried out with it. The valve gear has proved to be quite the most durable push-rod type yet seen by the Division. The mean effective pressure developed exceeds anything known to the Division in air-cooled cylinders. It is of $5^5/_8$ in. bore by $6^1/_2$ in. stroke and has a compression ratio of 54 to 1. When running at 1800 r.p.m. it regularly exerts 145 lb. per sq. in. brake mean effective pressure, and occasionally 155 lb. per sq. in.

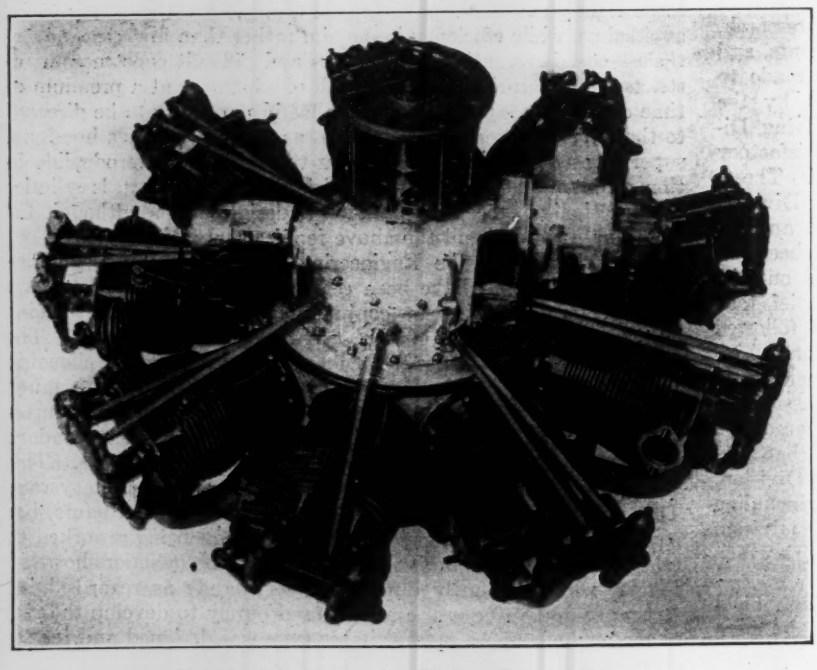

FIG. 1 9-CYLINDER AIR-COOLED RADIAL ENGINE

The air-cooled Liberty cylinder shown in Fig. 3 follows the same general construction as that of the type M, except that an overhead camshaft valve gear is used. A point of interest is the method of shrinking on the head clamp ring. The ring cannot be dropped on the cylinder barrel after the head is shrunk as on the type M, but has to be loosely attached to the

barrel while the head is shrunk. It cannot be heated for shrinking on by normal methods without heating the head and barrel. This is overcome by passing a heavy low-tension current through the ring. This heats it to the shrinking temperature in a few seconds. The current is obtained from a butt-welding machine.

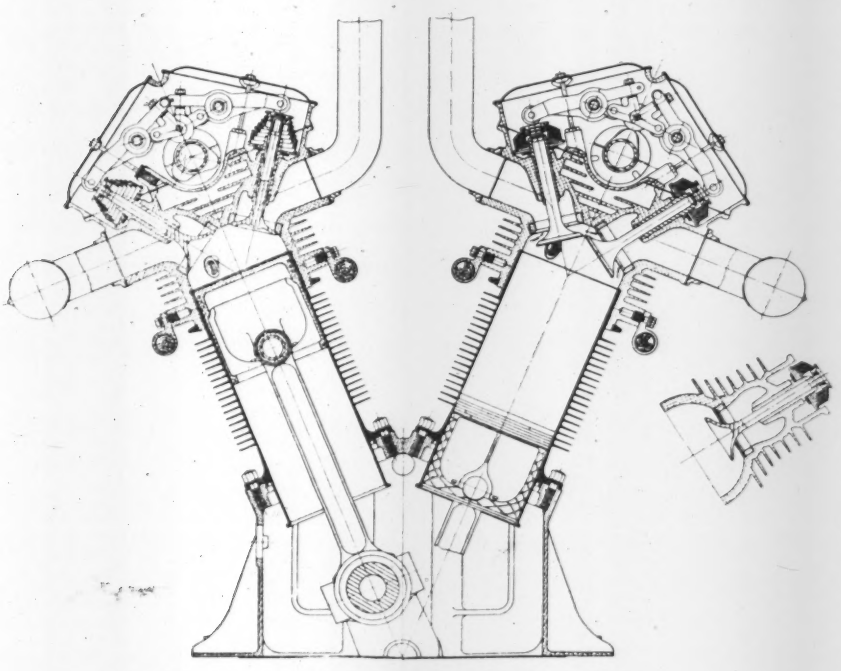

FIG. 3 CROSS-SECTION OF AIR-COOLED LIBERTY-12

The Engineering Division foundry has been responsible for the commercial production of aluminum cylinder-head castings. The castings are produced from stripper-plate patterns upon jolt-ram molding machines, and while these castings may appear difficult they are really much simpler and easier than an aluminum water-cooled cylinder casting.

An aluminum alloy consisting of 4 per cent copper, 2 per cent nickel, 1½ per cent magnesium, 91½ per cent aluminum (Fe 0.60 per cent max., Si 0.40 per cent max.), has been standardized for cylinder-head castings. This has remarkable strength at high temperature and produces very sound castings which do not warp in heat treatment. This alloy also gives remarkably good results in pistons and piston-pin bushings.

Generator Problems Due to Crankshaft Vibration

BY J. W. ALLEN,[1] DAYTON, OHIO

ONE CAN readily see how explosions can cause torsional displacements in the crankshaft of an airplane engine. Since the propeller is the flywheel of the engine, it can be considered as revolving at a constant velocity. This probably is not strictly true, but the variations are slight when compared to variations in the crankshaft itself. Such crankshaft "whips" are transmitted to all accessories.

In the Liberty-12 engine the generator drive shaft is geared directly to the non-propeller end of the crankshaft. The splined generator shaft fits into an internal spline of the generator drive. This connection is a slip fit. The generator turns one and one-half times crankshaft speed, there being one revolution of the generator for every four explosions. This means that the generator is subjected to four distinct "whips" in one revolution.

The problem of a dependable generator drive increases as a larger generator is installed. The size of armature increases with the generator size and the armature flywheel effect increases with the size of armature, and the stresses due to variation in angular velocity increase as the flywheel effect increases.

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In the early days of the Liberty-12 engine, it was found that the generator armatures failed frequently due to a shifting of the armature punchings on the shaft. This was overcome by providing the punchings with an internal spline and slipping the mover a splined generator shaft; but the stresses remained. It was determined at that time that the armature was subject to a total whip of 13 deg. Tests made on the 1-L generators (15 volts, 25 amperes) showed a whip of at least 10 deg. although later failures showed that a deflection of well over 15 deg. was present from the fact that the springs were deflected enough to gouge into the housing.

When the first 25-ampere generator was built a friction-clutch drive was incorporated, which was expected to slip during severe whips and thus protect the generator. Two difficulties arose: the clutch setting decreased with wear of friction surfaces, and the small driving shafts broke. When the difficulties became apparent a spring drive was incorporated. This construction permitted a strong driving shaft. Four sets of cantilever springs absorbed the angular variations. This first spring drive appeared satisfactory but failures began to crop up. Two changes were made. The cantilever-spring construction was changed from three 0.035-in. springs to eighteen 0.016-in. springs and the springs were so arranged that they would drag on the housing instead of gouging into it.

Tests made by the generator manufacturer showed that the heavy springs broke in half an hour, whereas the thin springs showed no failure in 90 hours under same test conditions. The weight of the 7-ampere armature is 3 lb., the 25-ampere armature, 10 lb., and the 50-ampere armature, 12.5 lb. At 2250 r.p.m. the flywheel energy in the armature mass is roughly 10 ft-lb. for the 7-ampere, 63 ft-lb. for the 25-ampere, and 110 ft-lb. for the 50-ampere generator. Assuming a 5-deg. angular variation between the armature and the generator drive, the energy that has to be

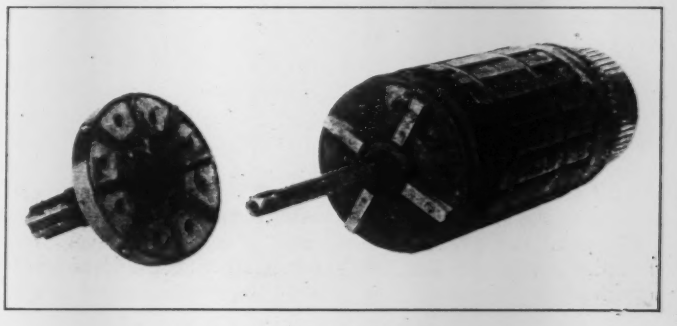

FIG. 1 RUBBER-CUSHION DRIVE

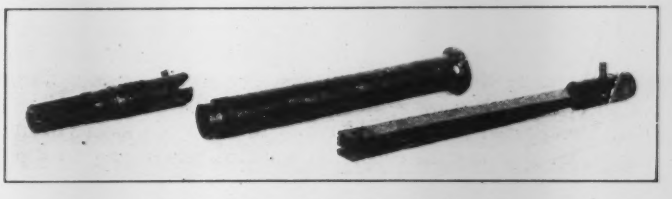

FIG. 2 TORSION-LEAF SPRING DRIVE

absorbed in the spring drive is 7.5 ft-lb. for the 25-ampere generator and 13 ft-lb. for the 50-ampere generator. This is applied at a radius of 1.5 in.

It is considered desirable to have a generator built that will withstand 500 hours on a Liberty-12 engine. At present 100 hours would be highly satisfactory. If such a generator were available, it would require no care except when the engine was being overhauled. The problem is complicated due to the fact that weight has to be a minimum.

The protective coupling must be light and compact. The present design using victrola-spring material shows promise. What

is really wanted is a coupling which will allow the armature to maintain a constant angular velocity, the coupling absorbing the variations of the pulsating drive. The present thin-spring drive is an approach to the desired constant-velocity condition.

At the present time two experimental couplings are under test; one a rubber-cushion drive and the other a torsion-spring drive. The rubber-cushion, Fig. 1, has not had sufficient engine time to permit of conclusions. Its advantages are that few parts can break and it can be readily repaired. The only argument against the rubber-cushion drive is wear on the rubber blocks, which is not considered vital. However, the service test being conducted should definitely prove or disprove its worth.

The torsion-leaf spring drive, Fig. 2, is very interesting. After 50 hours' engine time a locking pin was worn but did not fail. A flat key was put in and the generator is now undergoing further tests. The only disadvantage of this drive is that it increases the generator diameter slightly. It, however, permits of a shorter generator.

Preparations are now under way actually to measure the generator-drive deflection by locating a wheatstone-bridge network at the points where the amount of deflection is desired. The indicating portion is to be led to an oscillograph and the various deflections as recorded in the oscillogram will show the angular variation, when compared to a calibrating oscillogram which will be taken under conditions of no deflection. Should these tests come up to expectations the actual deflections and stresses in the generator drive will be better understood and the generator couplings designed accordingly.

Fatigue of Welds

BY R. R. MOORE,[1] DAYTON, OHIO

THE TENDENCY toward welded-steel-tube fuselages in airplane construction and the occurrence of several failures in service have augmented the importance of the reliable construction and subsequent inspection of welded joints and of a more adequate knowledge of the properties of such connections.

It is a comparatively simple matter to test a welded joint in tension with any of the standard static testing machines. A large number of such tests have been made, and they demonstrate in the case of low-carbon butt-welded tubes that the ordinary torch weld can be depended upon for a tensile strength of at least 80 per cent of the original strength of the tube. Higher values have been obtained. The arc weld gives similar results. There is little to choose in this respect. Although there is a large amount of such data available, we still are hardly in a position to make justifiable recommendations as to the strength of welds under service conditions.

The point to be brought out is that it is rarely, if ever, that a structure or machine part fails in service by a single application of a gradually increasing load as occurs when running a test in a tension machine to determine the ultimate strength of a material. On the contrary, the failure occurs after a number of applications of a load or of varying loads. The fuselage of an airplane is subject to considerable vibration due to both wind forces and engine vibration. These vibrations set up rapid alternations of stress in all of the parts of the fuselage. It is a matter of considerable importance, then, to know something about the resistance of welded joints to repeated stresses, and either to develop a means of overcoming the detrimental effects of repeated stresses, or to supply the designer with reliable information as to what he may expect from a welded joint under such conditions.

With these points in view, the Engineering Division of the Air Service is conducting an investigation on the resistance of welds to repeated stresses. The repeated-stress tests are being made by what is known as the rotating-beam method on a machine developed by the Division to handle 1-in-diameter tubes. Fig. 1 shows the general construction of this machine. It consists briefly of two main housings, in each of which runs a hardened-steel journal. The inside of the journal is fitted with a gripping device at the inner end of the housing which holds the ends of the tubular specimen. It is necessary to plug the ends of the tubes. Each housing is supported on pivot pins at the outer end so that the whole assembly is that of a simple beam supported at the ends on pivots. The beam is loaded at two points, as shown, each 10 in. from the pivot pins. The load is applied through knife edges set in a yoke which straddles the housing. The knife edges rest in seats attached to the housing. In order to facilitate the application of heavy loads, the hanger yokes are connected by hanger arms to a carefully constructed lever system which gives a 10 to 1 advantage. It is clear that under these conditions there is tension in the lower fibers of the tube and compression in the upper. Upon a rotation of 180 deg. the stresses are reversed. Upon a complete rotation the specimen is subjected to a complete cycle of stress, the maximum of value of which is determined from the loads at the load points by means of the common flexure formulas.

The results of the tests so far completed indicate that the resistance of either an arc or torch weld to repeated flexural stresses is very low as compared to the resistance of the tube itself. The endurance limit of both types of welds is practically the same—about 14,000 lb. per sq. in. See Fig. 2. The tests run to date, however, are not sufficient to determine the exact location of the endurance limit. They do show, however, that the endurance

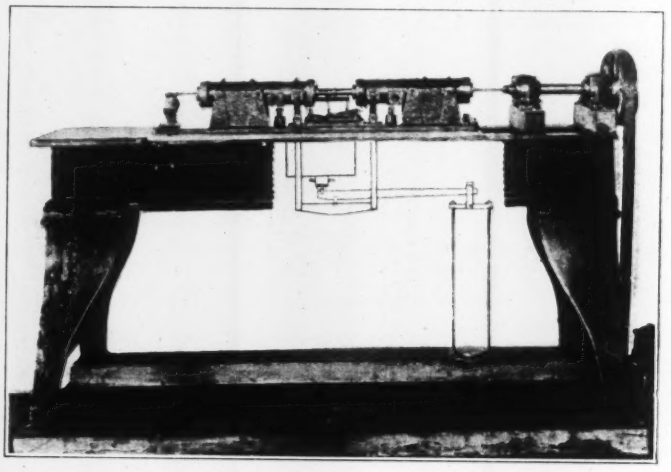

FIG. 1 FATIGUE MACHINE FOR TESTING TUBES

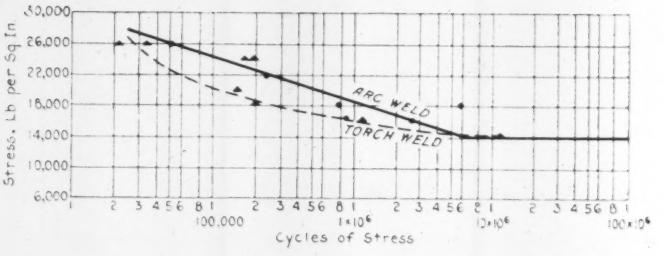

FIG. 2 ENDURANCE CURVES OF WELDED TUBES

limit is not greater than 14,000 lb. per sq. in. More complete tests may show a somewhat lower endurance limit, which of course makes the joint still weaker. Further tests are in progress to check this point. The endurance limit of a plain tube is about 28,000 lb. per sq. in., which is 100 per cent greater than the welded tube. Here is evidence, then, that a welded tube, under conditions similizing those found in service, is only one-half as strong. This surely is a vital consideration.

Another interesting point brought out by these tests is the location of the fracture of the specimen. In the static-tension test the torch-welded specimen breaks at a point about $\frac{1}{2}$ to 1 in. from the weld as shown in Fig. 3, due no doubt to the softening of the tube at this point by the heat of the flame, and also to the additional strength which the weld receives from a greater cross-section area due to the larger mass of the deposited metal at the weld. In the repeated-stress test, however, the fracture almost always occurs through the weld. The weld itself is evidently the weaker part of the unit under these conditions. Fractures of arc-welded tubes occur at the edge of the weld in both the static-tension and repeated-stress test. In this case the heating is local.

[1] Engineering Division, Air Service, McCook Field.

Further investigation into this subject is being made with a view to improving the strength of the weld by such methods as application of heat treatment, use of different materials, and modification of the design of the welded joint.

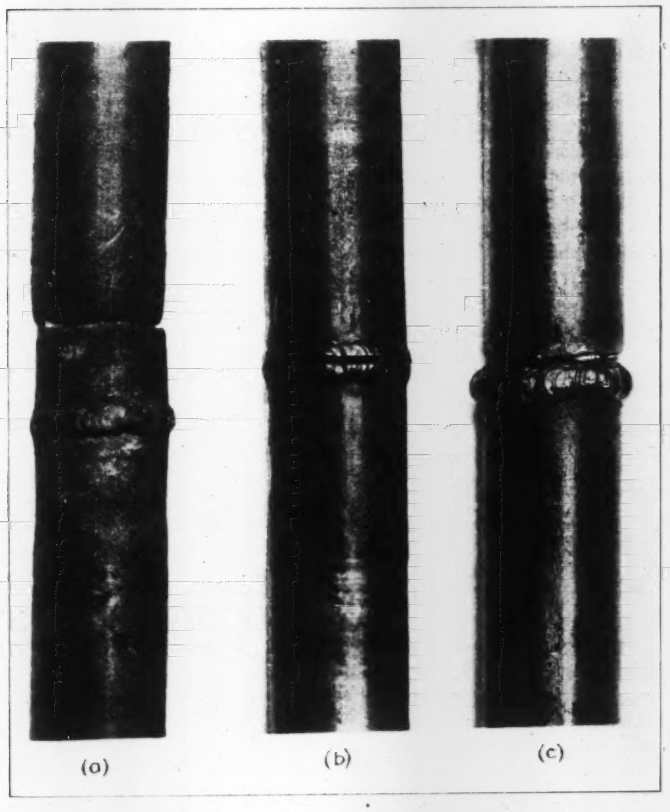

FIG. 3 FRACTURES OF WELDED TUBES
(*a*, Tension fracture of torch weld; *b*, fatigue fracture of torch weld; *c*, fatigue fracture of arc weld.)

Earth-Induction Compass

BY GEORGE P. LUCKEY,[1] DAYTON, OHIO

A MAGNETIC compass as ordinarily used on the ground is an accurate instrument, but if mounted on a vibrating board, subjected to rapid accelerations, and placed within a few feet of several hundred pounds of iron, its indications would not be of much value. In an airplane the compass must be mounted in front of the pilot where it can be easily visible. However, in front of the pilot are an engine, current-bearing wires, and machine guns, and beneath or behind him may be several thousand pounds of bombs in steel shells. In order that a compass card may set level its mass must be unbalanced to counteract the tendency of the north-seeking ends of the magnets to point downward. This makes the instrument subject to accelerational errors such as are due to vibrations or the turns and oscillations of the airplane. A magnetic compass also requires a certain time to assume the correct reading after a turn has been made, due to the period of the compass.

In spite of these difficulties the magnetic compass would give fairly reliable indications when flying straight and level were it not for the fact that the magnetic disturbance due to the presence of the huge mass of iron is continually changing, due to the effect of vibration and the change in the effect of the earth's field on different headings.

If it were possible to place the compass where disturbances of the earth's magnetic field were small, good results could be obtained, but in such a position it would not be visible to the pilot. Optical systems for reading at a distance have been tried and found impractical. A distant-reading magnetic compass has been built by Carl Bamberg of Germany. This employs a selenium cell as a means of transmitting the indication to a distance. However, in test this instrument has not given satisfactory results.

In an endeavor to obtain a satisfactory distant-reading compass the Engineering Division of the Air Service allotted funds to the Bureau of Standards for its development. In November, 1920, work was started on the induction type of a compass by Drs. Briggs and Hehl, and in 1921 the first satisfactory working model of an induction compass was completed and tested. As in any new field of endeavor, in spite of the large amount of development which has been done on this type of compass, much remains for the future. However, the models at present obtainable are in every respect, except durability, greatly superior to the magnetic compass.

In a magnetic compass a magnet is allowed to align itself with the earth's magnetic lines of force. In an induction compass a current is generated by rotating coils of wire in the earth's field and the direction of the field is determined by the amount of current generated at any instant in the various coils.

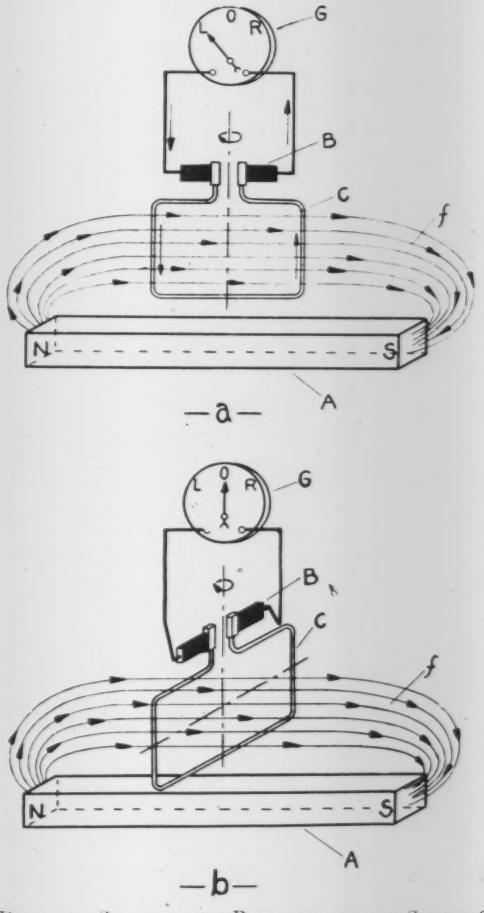

FIG. 1 DIAGRAMS SHOWING THE PRINCIPLE OF THE SINGLE-CIRCUIT INDUCTION COMPASS

To date two types of induction compasses have been developed, a single-circuit and a two-circuit type. In the single-circuit type a drum-wound armature equipped with a commutator similar to that used in a direct-current generator is rotated in the earth's field. (See Fig. 1.) This coil is placed where the magnetic disturbance is minimum and is rotated by a small propeller placed in the air stream. (See Fig. 2.) The current generated is taken to a small galvanometer placed in front of the pilot. The brushes are then rotated so that they are in contact through the commutator with a coil of wire in which no current is being generated and from the position of the brushes the direction of the earth's lines of force can be determined. The position of the brushes with reference to the axis of the plane will then give the direction in which the aircraft is heading. In practice the brushes are rotated by a controller placed on the instrument board through a rod going back to the generator. The controller is so marked that when it is set so that there is no deflection of the galvanometer pointer the course on which the airplane is headed can be directly read. If the airplane

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is turned to the right, current will flow through the galvanometer and the pointer will move to the right, and if the airplane is turned to the left the current generated will be in the opposite direction and the pointer will move to the left.

In the two-circuit type of induction compass two sets of stationary brushes are used placed 90 deg. apart. The current from the four brushes is led to the four corners of a wheatstone bridge placed on the instrument board. Through a rotating arm making contact with the bridge, current can be taken from any two opposite points

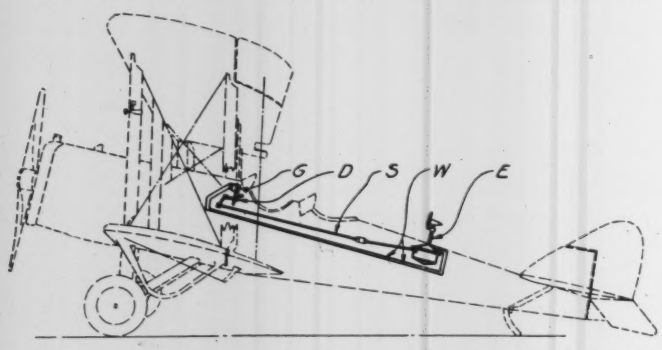

FIG. 2 INSTALLATION OF SINGLE-CIRCUIT INDUCTION COMPASS IN DH AIRPLANE

(*G*, galvanometer; *D*, controller; *S*, controller shaft; *W*, wires leading to galvanometer; *E*, compass.)

FIG. 3 SINGLE-CIRCUIT INDUCTION COMPASS WITH ACCESSORIES

to a galvanometer. The arm is rotated until no current is flowing through the galvanometer and the heading of the airplane can be read directly from a dial.

The generator is hung pendulously and the gyroscopic action of the rotating armature tends to stabilize it so that its axis remains vertical during slight oscillations of the airplane. (See Fig. 3.)

The induction compass has many advantages over the magnetic type of compass. It is more accurate. Five degrees is about the limit of accuracy obtainable with the present airplane magnetic compass. Using the present induction compass an accuracy of $1^1/_2$ deg. is obtainable. In many airplanes it is impossible so to place a magnetic compass that it will give even approximately accurate readings. Since it is distant-reading, an induction compass can be easily placed where it is undisturbed by local magnetic fields. The magnetic compass is affected by vibration and, once disturbed, requires considerable time to settle down. Vibration does not affect the induction compass and it instantly shows the course on which the airplane is headed.

Since to keep on the correct course it is only necessary to keep the pointer of the induction compass indicator on zero, it is easier to read than the magnetic compass. By the use of suitable relays it is possible to use the induction compass to operate mechanical devices. A recording induction compass has already been made and flight-tested by the Engineering Division. In this compass the energy from the generator was magnified by a relay to operate a pen moving on a rotating drum and an actual record of the course flown was obtained. By further development along this same line it should be possible to make the induction compass automatically steer the airplane on any desired course.

Magnesium and Its Alloys

BY SAMUEL DANIELS,[1] DAYTON, OHIO

AMONG the attributes of metallic magnesium are its very low specific gravity of about 1.738 (65 per cent of that of aluminum); its great affinity for oxygen and nitrogen, forming magnesium oxide at about 1110 deg. fahr. in air and the nitride at 1240 deg. fahr.; and its reactivity to water (and to mineral acids) with consequent corrosion to the carbonate or basic carbonate. The melting point of magnesium is 1204 deg. fahr. Its specific heat is higher than that of any other common metal. Its coefficient of expansion is slightly higher than that of aluminum. Its thermal conductivity may for low temperatures be taken as 0.35. The volume electrical conductivity of magnesium equals 38.5 per cent that of copper. Magnesium has a very high reflecting power in the visible spectrum.

Manufacture of Magnesium Ingot. Pure magnesium is prepared by the electrolysis of molten salts in an iron pot. In the United States the oxide and the chloride, and abroad the chloride, processes are used. The former employs a molten bath of magnesium oxide and fluorides, both non-hygroscopic. Magnesium is continuously plated out at the cathode and the oxygen burns to carbon dioxide at the anode, while the bath is constantly replenished with magnesium oxide. The molten electrolyte in the chloride process consists of a mixture either of anhydrous magnesium chloride and either potassium or sodium chloride, or of the dehydrated natural double chloride, carmallite ($MgCl_2.KCl.6H_2O$). In this method of extraction magnesium is plated out at the cathode, whereas the chlorine at the anode is removed by suction. The magnesium thus produced is usually purified by remelting and skimming. It is claimed that this process yields magnesium containing included chlorides from the electrolyte and from the action of the chlorine on the magnesium near the anode, which make the metal more susceptible to corrosion. In France, however, a modification of the chloride method has lately been developed which purports to free the metal from chlorides entirely.

Commercial Forms and Analysis of Magnesium Ingot. Magnesium metal is supplied in ingot, sheet, plate, tubing, rod, wire, ribbon, and powder. The grades of ingot offered by an American manufacturer follow:

Grade	Metallic Magnesium Content
No. 0	99.99 per cent (min.)
No. 1	99.85 per cent (min.)
No. 2	99.00 per cent (min.)

Grade No. 0 is an extra pure metal for special purposes; No. 1 is commercial pure metal, for structural purposes; and No. 3 is a grade suitable for deoxidizing or minor alloying, the impurities being mainly aluminum, iron, and silicon. The ingots are 12 in. long by $1^1/_8$ or $1^5/_{16}$ in. diameter, or 1 in. square, weighing, respectively, 0.75, 1.0, and 0.75 lb.

Both metallic and non-metallic impurities may be present in the metal. The metallic are aluminum, barium, calcium, iron, potassium, silicon, and sodium; the non-metallic are magnesium oxide, carbon, and such chlorides as those of calcium, magnesium, potas-

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sium, and sodium. The non-metallic class of impurities remain suspended in the metal and if hygroscopic, like magnesium chloride, absorb moisture and forms a saline solution which rapidly and progressively corrodes the metal and weakens it. Among the metallic impurities, potassium and sodium also act in this way. Corrosion is also caused by segregation of elements electronegative to magnesium, but this effect is lessened if these elements, such as aluminum, form solid solutions with magnesium. Iron is insoluble in magnesium; and calcium in small amounts has little effect, if any. Silicon may be present as Mg_2Si, which decomposes water. Magnesium oxide as dross is detrimental, as is also magnesium nitride, formed through decomposition of air during ordinary melting operations.

Wrought Magnesium and Its Uses. Pure magnesium metal is not generally used for casting purposes, but it may be worked (rolled, pressed, or extruded, spun, or drawn) at from 660 to 840 deg. fahr. to which temperatures the dies and tools must also be heated. Cold working of magnesium is not very successful. Sheet and plate may be obtained in a variety of gages, widths, and lengths, and in any temper from soft to full hard. Extruded tubing is produced in a number of tempers in outside diameters up to $1^1/_4$ in. and a thickness up to about $^1/_4$ in., but the tolerances must be greater than for drawn tubing in other metals. Rod and wire are extruded, and the latter is drawn to as small diameters as 0.005 in. Stranded cable is also available. The ribbon is rolled from fine-gage wire to thicknesses of 0.006 or 0.007 in. The powder for photographic and pyrotechnic purposes is produced by milling, etc., and supplied in 50- to 200-mesh grades.

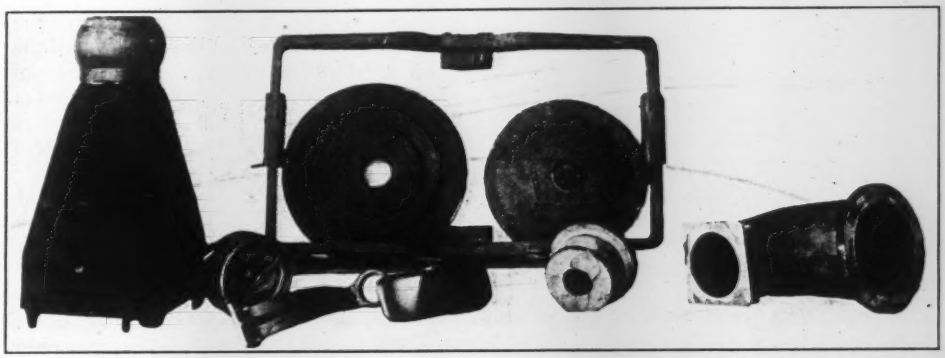

Fig. 1 Aircraft Castings in Magnesium-Base Alloys

The properties of pure magnesium are given in Table 1.

TABLE 1 MECHANICAL PROPERTIES OF PURE MAGNESIUM

Condition	Ultimate strength, lb./sq. in.	Proportional limit, lb./sq. in.	Compressive strength, lb./sq. in.	Shear strength, lb./sq. in.	Elong. in 2 in., per cent	Hardness Brinell 500 kg.	Hardness Scleroscope
Sand-cast	13,000	500	32,000	14,000	6	30	20
Extruded	28,000	1000	45,000	16,000	8	35	23
Rolled sheet	25,000	2500	.....	14,000	4	40	31
Annealed sheet	25,000	2000	.....	14,000	5	33	27

The modulus of elasticity in tension for magnesium and its alloys may be taken as about 6,000,000 lb. per sq. in. The proportional limit of magnesium and its alloys is quite low, a disadvantage when they are compared to aluminum and its alloys. Neither magnesium nor its alloys bend so well as aluminum and its alloys.

At elevated temperatures magnesium deforms fairly readily. From a strength of about 24,000 lb. per sq. in. at room temperature it falls to about 9600 lb. per sq. in. at 350 deg. fahr., and to 430 lb. per sq. in. at 1025 deg. fahr.

Various shapes of pure magnesium are commercially available. The pure metal will probably never assume much importance as a structural material.

Alloys of Magnesium

A variety of casting alloys prepared by American and foreign manufacturers and supplied in ingot form for remelting are available. These alloys may be prepared at atmospheric pressure, under a tight-fitting cover. Among the commercial alloys are the magnesium-aluminum series, containing from 4 to 12 per cent of aluminum, and sometimes small amounts of manganese or of silicon and calcium; the 99 magnesium-1 cadmium alloy; the 87 magnesium-13 copper alloy; and the elektron series, comprising the alloys containing varying quantities of aluminum and zinc, with small amounts of calcium. Magnesium alloys with small amounts of aluminum, copper, and zinc have been mentioned in the literature, which is not always authentic.

Magnesium-base alloys, with their lower melting points, do not oxidize or burn so readily as pure magnesium; but still, extreme care must be taken in melting and in pouring. Provision must be made for the peculiarities of the metal in patternmaking, the selection and doping of sand, in molding, gating, venting, core making, and in chilling. Some aircraft castings in magnesium alloys are shown in Fig. 1.

Melting Practice. The alloys may be melted in oil or gas-fired furnaces in cast-iron or steel crucibles. The furnaces of large capacity are of the tilting or preferably of the bottom-top type. When small crucibles are used, they are of the ordinary type or bottom or side-poured to tap sub-surface metal. The containers are closely charged with ingot and with or without clean scrap and the alloy is rapidly melted, preferably under cover. As the melting proceeds, more solid alloy may be added to the skimmed bath. Sometimes the metal is melted less conveniently and economically in a vacuum or under an inert gas. The temperature of the metal at no time should exceed 1335 deg. fahr. but should be carried to a point slightly above that to be used for pouring.

Use of Fluxes. As a general rule no flux should be added during melting, and stirring or excessive tilting is to be avoided. Sometimes sulphur is thrown on the metal during melting and again after skimming and before pouring, when the metal should be silvery. Black dross on the surface of the metal indicates overheating. Sub-surface dross is very difficultly separable from the molten alloy because of the slight difference between the two in specific gravity. The use of chloride fluxes is not to be recommended, as they are the very corrosive salts which the refining processes for the production of pure magnesium endeavor to eliminate.

Pouring Temperature and Rationale. The range of pouring temperature is from 1200 to 1300 deg. fahr., with 1250 deg. fahr. as the average for ordinary castings. Hotter metal is preferable to run thin sections and long distances. The lowest possible casting temperature should always be used and the temperature of pouring carefully controlled. The iron-constantan thermocouple is satisfactory for this work. The metal may be poured from the crucible used for melting, or from preheated crucibles of self-skimming iron ladles filled from the melting furnace. When it is necessary to melt two batches of metal in separate crucibles to pour off a large casting, a double-throw switch to the pyrometer enables closely parallel temperature regulation during melting, so that at the moment of pouring, the metal in both pots may be at the same temperature. After the metal has once attained the proper temperature for pouring, it should be poured immediately to prevent further oxidation and gas absorption. The skimmed metal should be poured rapidly into the mold. The skimmings may be quenched into oil. The magnesium alloys are not so hot-short as those of aluminum.

Cleaning of Pots. After the molten metal has been removed from the pot the latter is left for the skull to burn away. It is then heated up and cleaned hot; or after the skull has burned off it is filled with water and allowed to stand overnight. In the latter case cleaning is very easily accomplished.

Patterns. Patterns for magnesium-base alloys must embody some considerations which are not taken into account for aluminum-base alloys, although the shrinkage allowance for both is $^5/_{32}$ in. per ft. A pad from $^1/_8$ to $^3/_{16}$ in. thick should be added to the thickness of flat surfaces which appear in the cope and are to be machined. This precaution is necessary because flat surfaces in the cope are likely to include gas holes and dross skins.

Molding Sand. For green-sand molding a coarse sand, low in

clay content, and of good bonding power should be employed. Inasmuch as magnesium will burn in ordinarily tempered sand, it becomes necessary to mix well with the untempered sand some such substance as kerosene, naphthalene, glycerine, sulphur, etc., which will, when the metal flows into the mold, burn to an inert vapor which prevents the metal from igniting.

Green- and Dry-Sand Molding. The lightness of magnesium alloys makes it difficult to drive air and gases out of the mold and engenders slow flow. The mold should therefore be rammed lightly,

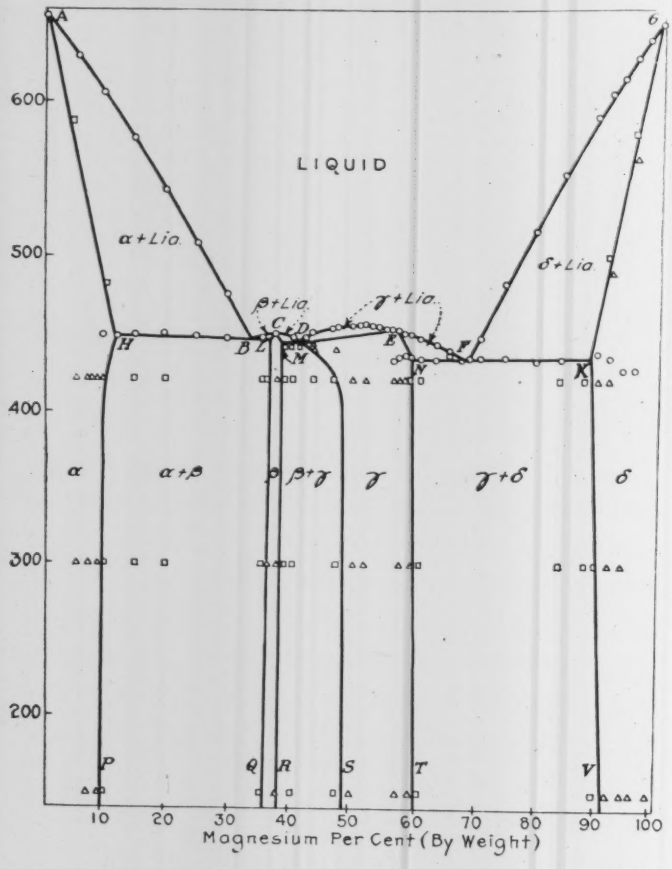

FIG. 2 EQUILIBRIUM DIAGRAM, MAGNESIUM-ALUMINUM SYSTEM (HANSON AND GAYLER)

vented freely, and poured rapidly. The mold should be reinforced carefully rather than hard-rammed to make deep lifts. The metal should not flow directly into the casting, but rather from runners to skim gate to actual entry runners that change the initial direction of flow. Risers are not necessary on small castings, but larger ones should be fed from a number of runners, and risers should be liberally used. Chills may be used at heavy sections. The pouring spure and skim gate must be kept full. It is sometimes advisable to tilt the mold or even to make provision for pouring vertically.

Dry-sand molds of coarse Albany or Sandusky sand need not be doped, as the moisture is removed by thorough baking. They should be warm to the touch when poured and should not be allowed to gather moisture. Neither should they be poured while too hot, for blows or oxidation from slow cooling may occur. Dry-sand molding is very commonly used.

Cores. Albany or Sandusky sand, with a binder, such as dextrin or water glass, which will not give off gases in the mold, may be used for cores. The cores are baked to remove the moisture. Water-glassed cores are rather difficult to file, but the sulphur-sand cores are quite satisfactory. The cores should crush easily, though the magnesium alloys are not so hot-short as those of aluminum.

Chill and Die Casting. Magnesium and its alloys may be chill-cast or die-cast with a plunger in heated metal molds.

Properties of Casting and Wrought Alloys. The magnesium casting alloys have a good range of mechanical properties, some cases may be substantially reinforced by working and by heat treatment.

Magnesium-Aluminum Alloys. Aluminum is added to magnesium in amounts up to 12 per cent and the series is the lightest of commercial alloys, the heaviest alloy as sand-cast having a specific gravity of 1.820. The alloys may be hot-worked and the cast alloys with more than 4 per cent of aluminum and the worked with more than about 6 per cent, may be heat-treated, with improvement of their mechanical properties.

The heat treatment of these alloys depends, like that of aluminum alloys, upon the solution of a soluble constituent, its retention in solid solution by quenching, and its subsequent precipitation in finely divided particles by artificial aging at elevated temperature. A glance at the equilibrium diagram by Hanson and Gayler in Fig. 2 shows that the delta solid solution is capable of dissolving about 10 per cent of aluminum as the gamma constituent, the compound Mg_3Al_2, at the eutectic temperature of 815 deg. fahr. (435 deg. cent.). The solubility of Mg_3Al_2 decreases with falling temperature, possibly somewhat more rapidly than the diagram indicates, according to an investigation of the Aluminum Company of America. This compound appears in the sand-cast alloys containing 4 per cent of aluminum and over.

The solution treatment is designed therefore to cause the solution and retention of Mg_3Al_2 in solution; and this is effected by soaking at about 800 deg. fahr., for short periods of time in the case of worked metal and for more extended periods in the case of castings, and subsequently quenching. The precipitation treatment follows, and is carried out at from 300 to 480 deg. fahr. for a number of hours and causes reprecipitation of Mg_3Al_2. As in the case of the precipitation treatment of aluminum alloys, the results from artificial aging may be varied by the adjusting both of the temperature

TABLE 2 MECHANICAL PROPERTIES OF MAGNESIUM-ALUMINUM ALLOYS

Composition	Density, grams per cc.	Lb. per cu. in.	Process of manufacture	Condition or heat-treatment	Tensile strength, lb. per sq. in.	Per cent elongation in 2 in.	Per cent reduction in area	Proportional limit, lb., per sq. in.	Lb. per sq. in. for 1/4 per cent extension under load	Compressive strength, lb. per sq. in.	Shear strength, lb. per sq. in.	Hardness: Brinell 500-kg. load, 10-mm. ball	Hardness: Shore scleroscope
96% Mg. 4% Al	1.767	0.0638	Casting (sand)	As cast	25,000	9	8	2,500	7,000	45,000	17,000	42	25
			Extrusion	As extruded	36,000	20	33	3,000	11,000	52,000	19,000	48	28
			Rolling (sheet	As rolled	41,000	8.5	10	7,500	10,500	....	19,000	61	38
			Rolling (sheet)	Annealed	34,000	16	15	3,000	8,500	....	19,000	46	32
94% Mg. 6% Al	1.780	.0643	Casting (sand)	As cast	24,000	7	7	2,000	8,500	46,000	18,000	46	28
			Casting (sand)	Heat treat.	25,000	5	5	2,500	9,000	46,000	20,000	50	31
			Extrusion	As extruded	41,000	16	21	5,500	15,000	58,000	22,000	54	32
			Extrusion	Heat treat	41,000	10	10	8,000	14,500	58,000	22,000	53	31
			Rolling (sheet)	As rolled	46,000	3	5	5,000	14,500	....	22,000	75	43
			Rolling (sheet)	Annealed	38,000	9.5	9	4,000	11,500	....	19,000	52	30
92% Mg. 8% Al	1.790	.0647	Casting (sand)	As cast	20,000	2	2	2,000	9,000	45,000	18,000	53	31
			Casting (sand)	Heat treat.	27,000	5	5	3,000	8,500	47,000	20,000	52	31
			Casting (sand)	Heat treat.	30,000	1	1	5,000	12,000	53,000	22,000	73	43
			Extrusion	As extruded	46,000	9	12	10,500	15,500	61,000	22,000	59	37
			Extrusion	Heat treat	47,000	4	4	8,000	15,000	64,000	23,000	68	38
			Rolling (sheet)	As rolled	46,000	7	7	11,000	15,000	....	22,000	69	40
			Rolling (sheet)	Annealed	41,000	8.5	8	5,000	14,000	....	19,000	54	31
90% Mg. 10% Al	1.806	.0652	Casting (sand)	As cast	19,000	1	1	2,500	10,000	49,000	19,000	57	33
			Casting (sand)	Heat treat.	23,000	2.5	3	3,500	9,000	48,000	21,000	55	29
			Casting (sand)	Heat treat.	27,000	1	1	10,000	13,000	53,000	25,000	76	43
			Extrusion	As extruded	47,000	8	10	7,000	14,000	66,000	24,000	68	38
			Extrusion	Heat treat.	55,000	2	2	9,000	15,000	62,000	24,000	83	50
			Rolling (sheet)	As rolled	48,000	5	5	4,000	13,000	....	21,000	71	41
			Rolling (sheet)	Annealed	44,000	3.5	3.5	7,000	14,500	....	21,000	70	40
88% Mg. 12% Al	1.820	.0657	Casting (sand)	As cast	19,000	0.5	None	4,500	11,000	47,000	18,000	65	44
			Casting (sand)	Heat treat.	24,000	1	1	4,500	10,000	49,000	21,000	64	42
			Casting (sand)	Heat treat.	31,000	0.5	None	12,000	16 000	56,000	22,000	94	64
			Extrusion	As extruded	45,000	7.5	8	10,500	15,000	72,000	24,000	71	46
			Extrusion	Heat treat.	45,000	1	1	13,000	16,500	67,000	27,000	90	55
			Rolling (sheet)	As rolled	53,000	3	2	16,000	16,000	....	24,000	81	47
			Rolling (sheet)	Annealed	45,000	3	3	8,000	15,500	....	22,000	82	47

and of the period. In general, the effect of artificial aging is to increase the strength, proportional limit, and hardness and to decrease the percentage of elongation as referred to the properties proceeding from the solution treatment. Some of the alloys higher in aluminum undergo alteration of certain of their properties through the artificial aging treatment alone.

Table 2, from the handbook on Magnesium by the American Magnesium Corporation, concerns the magnesium-aluminum series. The addition of aluminum up to 4 per cent increases the ultimate strength, ductility, and hardness of the cast alloys. Extrusion of the 4 per cent alloy more than doubles the percentage (to 20) of elongation. Further additions increase the hardness, with loss of ductility, with favorable effect on the strength only of the worked materials. The addition of aluminum increases the proportional limit; and this effect is enhanced by the addition of manganese.

One of the casting alloys used by the Air Service to meet minimum requirements of 20,000 lb. per sq. in. ultimate strength and 4 per cent in elongation contains 4 per cent of aluminum and a small amount of manganese. This alloy has been sand-cast for crankcases, supercharger casings, instrument covers, camera mounts, etc. Its properties as sand-cast have averaged 23,000–7.0–45 and as machined 19,500–5.0–45.

The structure of this alloy as sand-cast is shown in Fig. 3. The constituents are Mg_3Al_2, an iron-bearing compound, Mg_2Si, and possibly manganiferous particles.

Experiments have been conducted with several other of the alloys in aircraft construction. The 8 per cent aluminum alloy has been worked into supercharger impellers and propellers. The 10 per cent aluminum, the 12 per cent aluminum, and the 8 aluminum-1 cadmium-1 copper alloys have been chill cast for pistons; and the 10 aluminum-1 silicon alloy has been used for piston-pin bushings.

Magnesium-Cadmium Alloys. Cadmium has been commercially added to magnesium in small amounts, and the alloys have been used in the cast and rolled form.

Magnesium-Copper Alloys. Though the 10 per cent copper alloy has been somewhat used, the 87 magnesium-13 copper alloy has enjoyed considerable prominence through its successful application in Germany as a piston material. The strength of the latter alloy at moderately elevated temperatures is superior to that of the 96 magnesium-4 aluminum alloy. In aircraft work chill-cast pistons of this alloy have operated without erosion, corrosion, or excessive wear; and it seems satisfactory for piston-pin bushings, though not for main or connecting-rod bearings. It has also been used for pulleys in lighter-than-air craft. The alloy is also furnished in the extruded condition. Its properties as sand-cast are 20,000–1.5–55. Its specific gravity is about 1.97. The structure consists essentially of the hard eutectic of Mg_2Cu and Mg in the magnesium-rich solid solution (Fig. 4). The presence of the hard compound in the soft matrix gives the alloy its good bearing qualities.

Magnesium-Aluminum-Zinc Alloys. The electron alloys are suitable for general purposes in the foundry, and meet the previously mentioned tensile requirements of the Air Service. The extruded and rolled electron alloys are said to have tensile properties similar to those of the magnesium-aluminum alloys. The structure of one of the cast alloys is to be seen in Fig. 5. The hard constituents are the grayish white eutectic of $MgZn_2$ and Mg, and two impurities, the blue Mg_2Si and another blue-gray constituent, probably containing iron.

SUMMARY

The magnesium-base alloys are characterized by a wide range of mechanical properties. Their lightness and ready machinability make them available for a number of commercial uses. They are on the other hand costly, difficult to cast, quite corrodible, and low in proportional limit and in tension modulus. The factor of high cost,[1] however, is only a transient condition, for it is only a matter of time until magnesium will be sold on such a basis that its price will compare favorably, volumetrically, with that of aluminum. As to the casting of magnesium or its alloys, foundry difficulties may be largely eliminated by attention to the inherent characteristics of the material. Progress in this matter and with respect to prevention or mitigation of corrosion is to be expected. The design of magnesium-base castings of course should take into account the mechanical properties of the materials as with alloys of other metals.

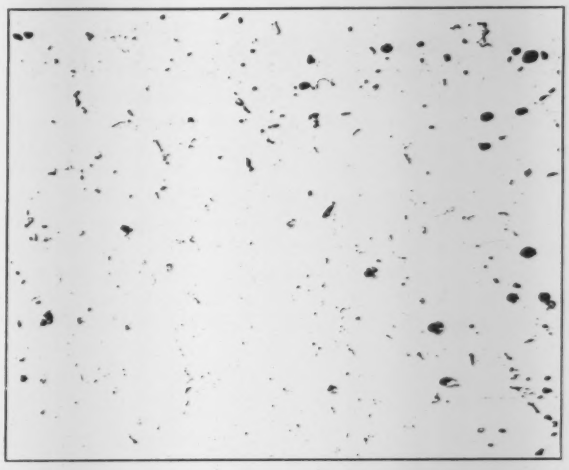

FIG. 3 MAGNESIUM-ALUMINUM-MANGANESE ALLOY (×100)

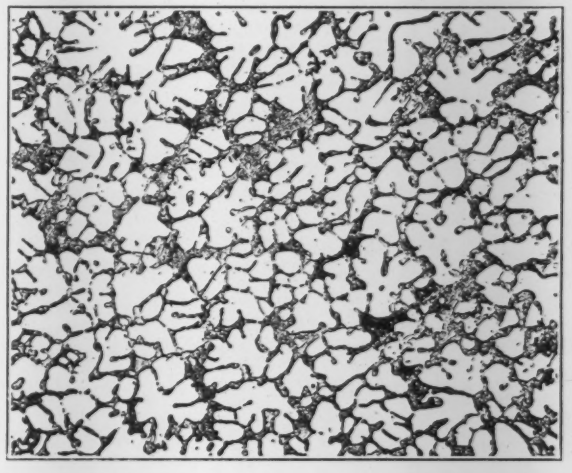

FIG. 4 MAGNESIUM-COPPER ALLOY, SAND-CAST (×100)

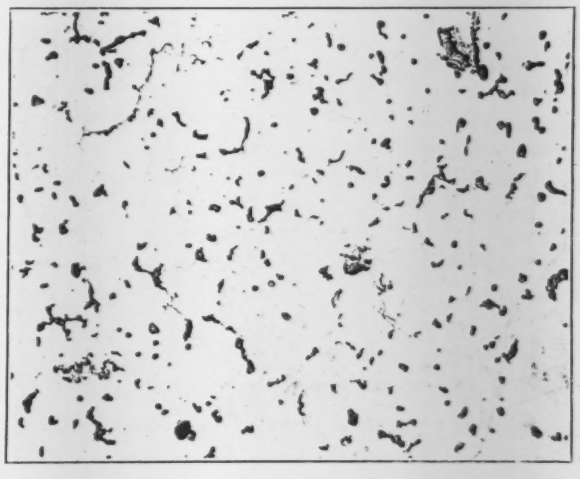

FIG. 5 MAGNESIUM-ZINC-ALUMINUM ALLOY, SAND-CAST (×100)

ACKNOWLEDGMENT

The author gratefully acknowledges the helpful criticism and data furnished to him by the American Magnesium Corporation, Niagara Falls, N. Y. and by the Elektron Metals Corporation, New York, N. Y.

[1] The present prices follow:
Magnesium ingot (99.85) in 1000-lb. lots, $1.00 per pound
Magnesium alloy ingot in 1000-lb. lots, $1.14–1.25 per pound
The prices for sheet, tube, forgings, rod, etc., will be quoted by the manufacturers upon application.

A Review of Steam-Turbine Development

By HANS DAHLSTRAND,[1] MILWAUKEE, WIS.

The paper is chiefly concerned with the effects which the use of steam at higher pressures and temperatures has had upon the efficiencies of steam turbines and upon the materials used in the construction of their parts. The use of high-back-pressure steam turbines, to be installed in connection with boilers of higher pressure in such a way as to exhaust into the steam mains and turbines of existing plants, is carefully analyzed for units of 10,000- and 30,000-kw. capacity, and the effects on the efficiency are graphically presented. A discussion of materials for use with the higher pressures and temperatures is included, with some curves showing the results of investigations made by the Allis-Chalmers Co. on various metals and the characteristics which should be possessed by a metal suitable for use at 1000 deg. fahr. The paper also contains some comments on corrosion and erosion of turbine blades, and shows the effects of these destructive forces on blades of different composition.

AMONG the various prime movers employed in producing electrical energy the steam turbine develops by far the largest amount of power used, and on this account is receiving much attention from engineers. But while great improvements have been made, there are still great possibilities for further improvements, both in design and in application. This paper is an attempt to indicate a few of these possibilities.

Present Maximum Standard Steam Temperature and Pressure

The maximum steam temperature which may be regarded as commercially practicable at present is about 700 deg. fahr., with temperatures ranging from 725 to 750 deg. fahr. for limited periods. The maximum standard pressure is generally about 350 lb. gage at the turbine with many installations contemplating pressures of 385 lb. gage.

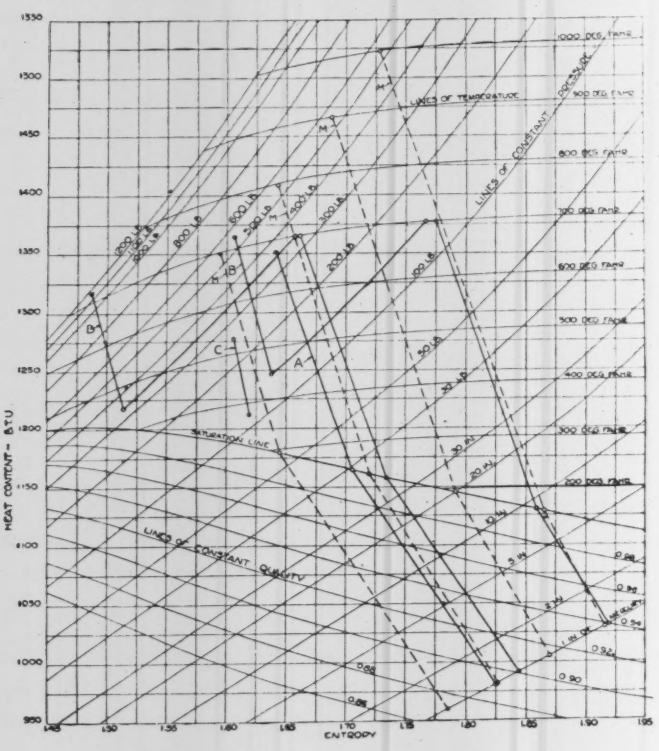

FIG. 1 VARIOUS TYPES OF EXPANSION LINES

Turbines for these steam pressures and temperatures have been in successful operation in many plants; and for sizes up to 50,000 kw. it appears that the turbine for 350 lb. gage and 700 deg. fahr. will, in nearly all cases, be of the single-cylinder, single-exhaust-nozzle type. The general efficiency characteristics and expansion through such a turbine is shown as curve A on the Mollier diagram, Fig. 1.

Developments during the past several years have indicated the feasibility of employing steam pressures of 500 to 600 lb. gage, with temperatures in the 700-deg.-fahr. field. In this country two new plants of these general characteristics have been put into successful operation. These plants are designed to accomplish eventually reheating at a pressure of 120 lb. gage after expansion from 550 lb. gage, the steam temperature at this pressure being raised again to 700 deg. fahr., and the expansion completed to a vacuum slightly greater than 29 in.

The turbine arrangement for this type of plant has a greater divergence than is the case for the 350-lb. units. For instance, the two 550-lb. plants in this country contain single-cylinder, two-cylinder, and three-cylinder turbines, with both single and double flow of the steam. The general characteristics of efficiency and expansion for a turbine operating on the reheat cycle outlined in the foregoing is shown as curve B on the Mollier diagram, Fig. 1. This particular expansion line was drawn from estimates made for a two-cylinder, two-generator, cross-compound 50,000-kw. unit.

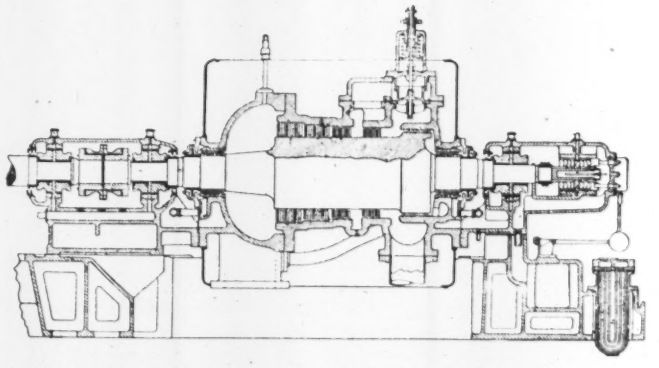

FIG. 2 5000-KW., 1800-R.P.M. STEAM TURBINE, 250 LB. GAGE STEAM PRESSURE, 125 DEG. FAHR. SUPERHEAT, 100 LB. GAGE BACK PRESSURE

High-Back-Pressure Steam Turbines

In plants using large quantities of steam for heating and process work at comparatively high pressures it would be economical to install steam turbines receiving steam from the boilers at very high pressures and exhausting against the back pressure required. The gain to be obtained from such a turbine installation has until recently been largely prevented because no standard boilers for these higher pressures have been available.

However, there has been at least one pioneer application of this type of apparatus. An Akron rubber mill using large quantities of steam at about 100 lb. pressure in connection with the manufacturing process, found it possible, in spite of a relatively low initial boiler pressure, 250 lb. gage, to install a turbine between the boiler header and the pipe supplying the 100-lb. process steam.

The turbine was of 5000-kw. capacity, operating at 1800 r.p.m. and regulated as to load by a back-pressure regulator and a standard centrifugal type of governor. This unit has confirmed the opinion of the author that the highest turbine efficiency would be obtained in the high-pressure sections, or elements, of large steam turbines.

Fig. 2 shows a section through a turbine unit of the type described. The efficiency characteristics of the unit are shown as curve C on the Mollier diagram, Fig. 1. Observations made during regular operation show that the turbine efficiency ranges between 75 and 85 per cent.

High-Back-Pressure Turbines in Existing Power Plants

In view of the successful application of high-back-pressure turbines the author has considered the possibility of applying high-

[1] M.E., Steam Turbine Engrg. Dept., Allis-Chalmers Mfg. Co. Mem. A.S.M.E.

Contributed by the Power Division and presented at the Spring Meeting, Milwaukee, Wis., May 18 to 21, 1925, of THE AMERICAN SOCIETY OF ENGINEERS. Abridged.

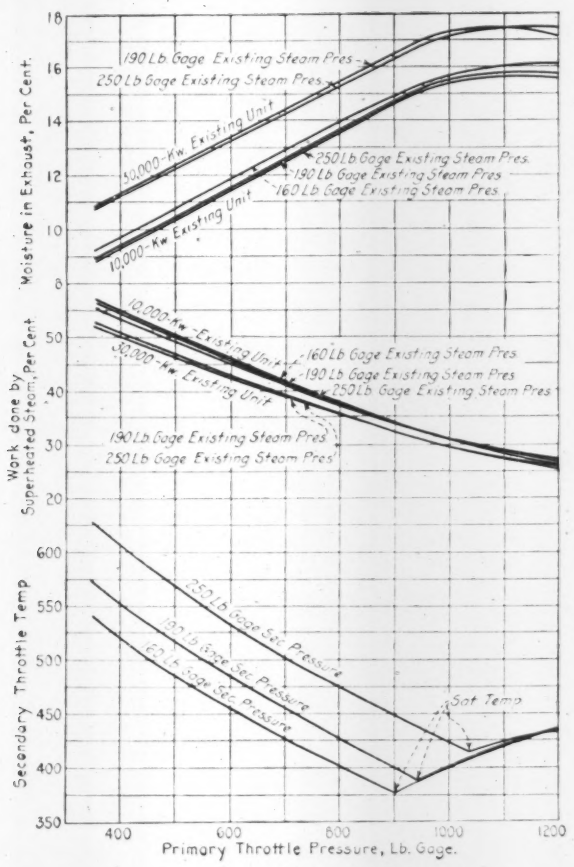
190 Lb. Gage Existing Steam Pres.
250 Lb Gage Existing Steam Pres.
50,000-Kw. Existing Unit
10,000-Kw. Existing Unit
250 Lb Gage Existing Steam Pres.
190 Lb Gage Existing Steam Pres.
160 Lb Gage Existing Steam Pres.
Moisture in Exhaust, Per Cent.
Work done by Superheated Steam, Per Cent
Secondary Throttle Temp
250 Lb Gage Sec. Pressure
190 Lb Gage Sec. Pressure
160 Lb Gage Sec. Pressure
Sat. Temp.
Primary Throttle Pressure, Lb. Gage.

FIG. 3 CHARACTERISTIC STEAM CONDITIONS OF EXISTING STEAM PRESSURES WITH VARIATIONS IN PRIMARY THROTTLE PRESSURES AT CONSTANT TOTAL STEAM TEMPERATURE OF 700 DEG. FAHR.

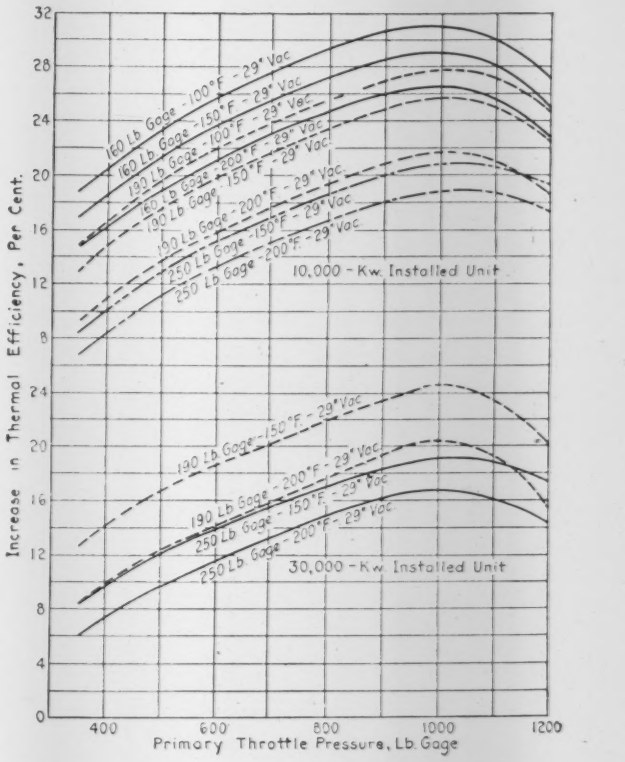
160 Lb Gage - 100°F - 29" Vac.
160 Lb Gage - 150°F - 29" Vac.
160 Lb Gage - 200°F - 29" Vac.
190 Lb Gage - 100°F - 29" Vac.
190 Lb Gage - 150°F - 29" Vac.
190 Lb Gage - 200°F - 29" Vac.
250 Lb Gage - 150°F - 29" Vac.
250 Lb Gage - 200°F - 29" Vac.
10,000 - Kw. Installed Unit
30,000 - Kw. Installed Unit
Increase in Thermal Efficiency, Per Cent.
Primary Throttle Pressure, Lb. Gage

FIG. 4 INCREASE IN THE EFFICIENCY OF THE COMBINED UNIT OVER THE EXISTING UNIT AT PRESENT STEAM PRESSURES AND TEMPERATURES, BASED ON VARIOUS PRIMARY THROTTLE PRESSURES AND TOTAL STEAM TEMPERATURE OF 700 DEG. FAHR.

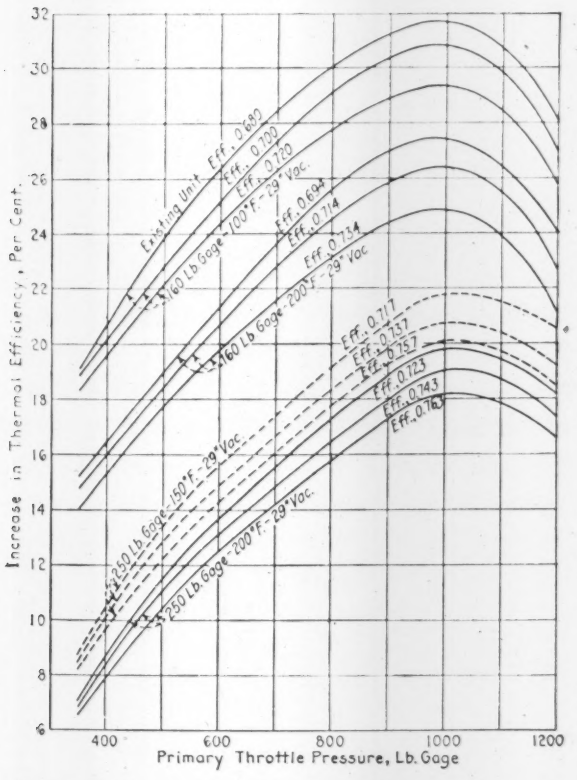

FIG. 6 INCREASE IN THE EFFICIENCY OF THE COMBINED UNIT OVER THE EXISTING UNIT AT PRESENT STEAM PRESSURES AND TEMPERATURES WITH VARIATION IN THERMAL EFFICIENCY OF EXISTING UNIT, BASED ON VARIOUS PRIMARY THROTTLE PRESSURES AND TOTAL STEAM TEMPERATURE OF 700 DEG. FAHR.

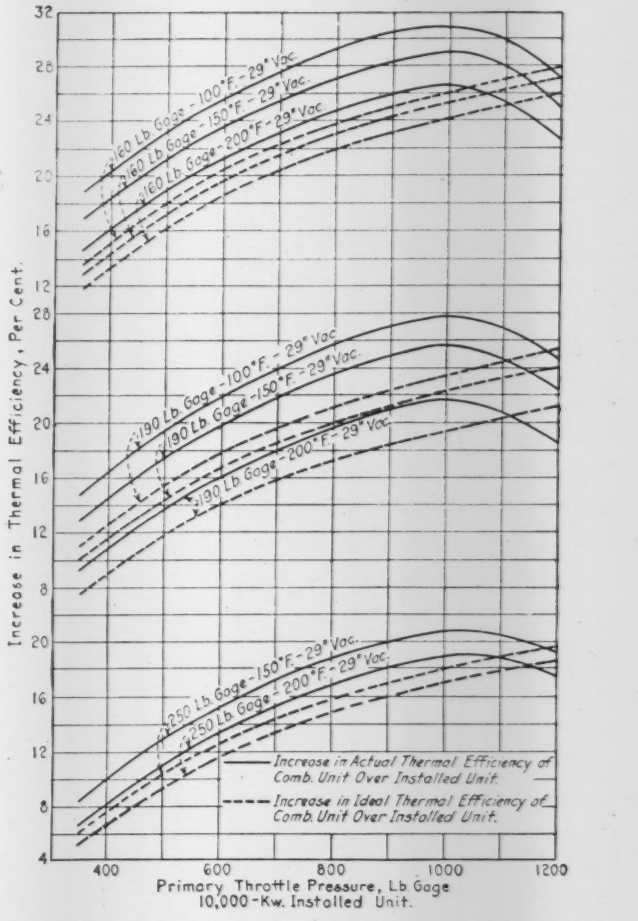
160 Lb Gage - 100°F - 29" Vac.
160 Lb Gage - 150°F - 29" Vac.
160 Lb Gage - 200°F - 29" Vac.
190 Lb Gage - 100°F - 29" Vac.
190 Lb Gage - 150°F - 29" Vac.
190 Lb Gage - 200°F - 29" Vac.
250 Lb Gage - 150°F - 29" Vac.
250 Lb Gage - 200°F - 29" Vac.
Increase in Actual Thermal Efficiency of Comb. Unit Over Installed Unit.
Increase in Ideal Thermal Efficiency of Comb. Unit Over Installed Unit.
Increase in Thermal Efficiency, Per Cent.
Primary Throttle Pressure, Lb Gage
10,000-Kw. Installed Unit.

FIG. 5 INCREASE IN THE EFFICIENCY OF THE COMBINED UNIT OVER THE EXISTING UNIT AT PRESENT STEAM PRESSURES AND TEMPERATURES, BASED UPON VARIOUS PRIMARY THROTTLE PRESSURES AND TOTAL STEAM TEMPERATURE OF 700 DEG. FAHR.

pressure units in the remodeling of existing plants. The problem would be worthy of detail solution if a high-efficiency, lower-pressure, existing plant were considered as being obsolete only in respect to the boiler-room pressure and temperature. The remodeling would then consist of the installation of new boiler-room equipment operating at a higher pressure and temperature, and a high-back-pressure turbine taking steam from the boilers and exhausting into the older and lower-pressure header system.

The author has made certain efficiency comparisons which may be of service in considering an estimate for remodeling older plants. For the general cases considered, the new boiler pressure, which has been called the primary pressure, might be 550 lb. gage or greater, and the steam temperature about 700 deg. fahr. The proposed high-back-pressure element would be designed to exhaust at the original boiler pressure. No account has been taken of the gain to be obtained by the possibility of using extracted steam for feedwater heating for the older or existing units.

The efficiency of steam turbines is affected to the greatest extent by the percentage of the total work of the turbine which is done by superheated steam. The greater the percentage of work done by superheated steam, the less will be the mean percentage of moisture in that part of the turbine using saturated steam. Fig. 3 shows the effect of primary throttle pressure on the percentage of work done by superheated steam, the moisture in the exhaust steam, and the temperature of the steam exhausted from a high-back-pressure steam-turbine element for the two different sizes of turbine units, 10,000 kw. and 30,000 kw., and for several secondary or existing steam pressures. It will be noted that for a primary throttle pressure of 550 lb. gage the secondary throttle temperature and the moisture in the exhaust steam cover the normal range for the older high-efficiency plants; that is, the secondary throttle temperature from 475 deg. at 160 lb. gage to 550 deg. at 250 lb. gage. The analysis covers 190- and 250-lb. gage steam pressures for 10,000-kw. and 30,000-kw. units with an additional steam pressure of 160 lb. gage for 10,000-kw. units.

Fig. 3 shows generally that for the highest pressure for which boilers are commercially available, 550 lb. gage, the remodeling of a boiler room and the installation of a high-back-pressure turbine would introduce no operating conditions which would cause an average existing turbine unit to suffer a loss in economy.

Taking full account of those factors which influence turbine efficiency, calculations have been made to show the increase in thermal efficiency which might be expected if higher boiler pressures and high-back-pressure turbines were applied to existing plants, using units of 10,000- and 30,000-kw. capacity. The range of pressures has extended to 1200 lb. gage, the primary initial steam temperature being, for all pressures, about 700 deg. fahr. The calculations have been based on expansion in the high-back-pressure turbine to different exhaust pressures ranging from 160 to 250 lb. gage. The results of these calculations are summarized graphically in Fig. 4. The increase in thermal efficiency falls off in the higher pressure range above 1000 lb. gage. In general it may be said that this decrease in the gain to be expected results from the greater proportion of the work which is done by moist steam when the pressure is increased and resuperheating is not resorted to.

The increase in thermal efficiency for different primary throttle pressures as approximately estimated for actual turbine installations checks in general characteristics with that indicated by theory for prime movers of 100 per cent efficiency. As a matter of comparison, the probable actual gain and the theoretical gain for the use of higher steam pressures and high-back-pressure turbines with an existing plant have been summarized graphically in Fig. 5. In this diagram the actual gain is, for the most part, greater than the theoretical, a condition which is obvious when it is considered that the inefficiency of the lower-pressure existing machine will increase the gain to be made.

Mention has been made of the fact that the inefficiency of an existing plant will increase the gain to be expected over that indicated by theory if the existing turbine is combined with a high-back-pressure turbine as a unit. The estimates and calculations have been made for 10,000- and 30,000-kw. units, the chief difference between these being a difference in the assumed efficiency. Calculations showing the effect of the lower-pressure or existing turbine's efficiency on the gain to be expected from the higher-pressure combination under discussion have been made for a limited range of turbine efficiency, 4 per cent, and for steam pressures from 160 to 250 lb. gage at different superheats. The results of these calculations are shown in Fig. 6.

HIGHER-TEMPERATURE DEVELOPMENT

The percentage of work done by superheated steam for a given condition has a marked effect on the average moisture in the steam in the lower-pressure stages of a turbine. As is also well known, if the steam temperature is increased at constant pressure there will result a marked decrease in the moisture found in the steam exhausted from the turbine. Tests and theory have likewise demonstrated that the efficiency of the turbine is greatly affected and benefited by a reduction in the amount of work done by moist steam.

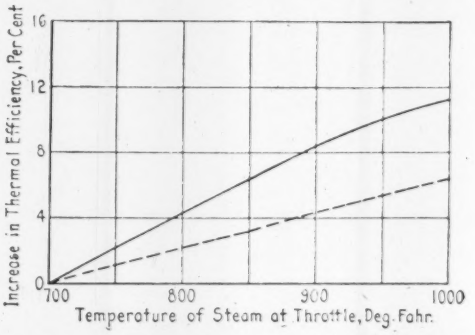

FIG. 7 INCREASE IN THERMAL EFFICIENCY AT CONSTANT PRESSURE WITH INCREASING TEMPERATURE, BASED ON 550 LB. GAGE AND 700 DEG. FAHR.

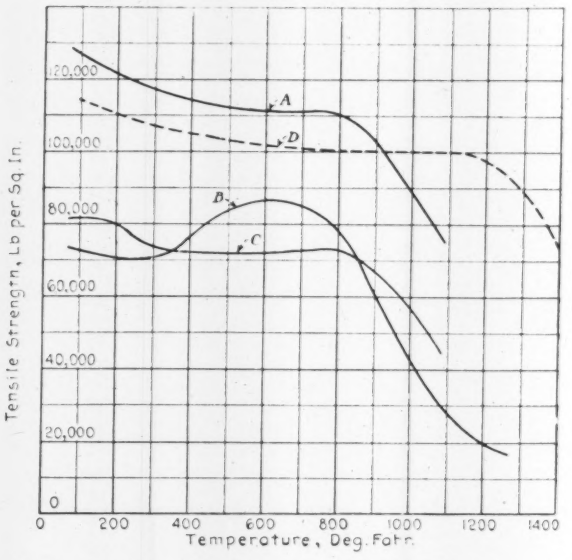

FIG. 8 PROPERTIES OF METALS AT HIGH TEMPERATURES

A Rolled high-chrome, high-nickel steel
B Forged 3½ per cent nickel steel
C Forged monel metal
D Metal suitable for higher temperatures

It follows, therefore, that the theoretical increase in thermal efficiency which occurs for an increase in initial temperature will be exceeded in the performance of an actual machine.

Taking the effect of moisture on the stages of a turbine at a value which has been confirmed by tests, the increase in thermal efficiency which will most probably be obtained if steam at a pressure of 550 lb. gage is increased in temperature from 700 to 1000 deg. fahr. is shown by the upper curve of Fig. 7. The lower dotted line of Fig. 7 shows the theoretical increase in thermal efficiency.

The form of the expansion lines and the efficiency characteristics for the different turbines considered, including those operating at proposed temperatures of 1000 deg. fahr., are all designated by the letter M on the Mollier diagram, Fig. 1. Curve D of Fig. 1 shows an expansion line worked up from tentative estimates made for a 1200-lb., high-back-pressure, proposed turbine installation, and it is felt that it represents fairly well the efficiency that

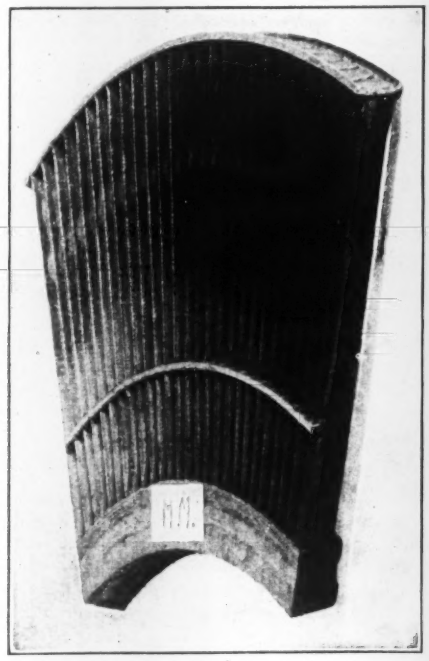

FIG. 9 EFFECT OF EROSION ON TURBINE BLADING
(Monel metal.)

FIG. 10 EFFECT OF EROSION ON TURBINE BLADING
(3½ per cent nickel steel.)

FIG. 11 EFFECT OF EROSION ON TURBINE BLADING
(80 per cent copper, 20 per cent nickel.)

may be expected with a machine designed for these conditions and not to function primarily as a reducing valve.

EFFECT OF MOISTURE IN SATURATED STEAM

Moisture is a factor which seriously limits the attainable efficiency of a steam turbine. For turbines operating in plants having superheated steam the moisture is formed as the result of work being done, and the greater the efficiency, the greater the moisture at exhaust. From these considerations it appears that the best remedy would be a device which would remove the moisture at the exhaust of one stage so that it would not be added to the stages following. Such an arrangement would probably result in a gain, because it is also known that the loss in efficiency in a given stage increases with the percentage of moisture.

The author believes that the present construction of turbines does not offer those characteristics which have resulted in successful separation and removal of moisture in other apparatus. The most effective separating devices have uniformly depended on a low velocity of the steam and, in some cases, the employment of large surfaces in combination with low velocity. These two features are not practicable in the present designs, in which commercial considerations require the maximum possible number of low-pressure stages in a single cylinder.

It is true that the flow of steam through the lower-pressure stages concentrates the moisture near the periphery of the turbine blading, but the same observations have also shown that the cylinder wall of the turbine is not covered with a continuous film of water. The author believes, therefore, that different methods of extracting steam at interstage points will have but a slight effect on the efficiency of the low-pressure stages of a given turbine. Further, there seems to be no immediate prospect of the application of effective moisture-separating features becoming commercially feasible.

FIG. 12 EFFECT OF EROSION ON TURBINE BLADING
(Ordinary carbon steel.)

AUXILIARY TURBINES FOR POWER-PLANT SERVICE

The subject of a reliable and economical drive for the auxiliary apparatus is still in the controversial stage. Opinions differ greatly, with some indication that the small turbine for this service may again find favor. The use of the electric motor for auxiliary drive has no doubt been extended through the rapid development of feedwater heating by interstage extraction of steam from the main turbine and to a lesser extent by the use of direct-driven auxiliary generators. These two arrangements not only increased the application of the electric motor to auxiliary service but tended to supplant the steam-driven house generator set as a source of power for the electrically driven auxiliaries.

The smaller-size turbine for direct connection to the larger and essential auxiliaries can be vastly improved both as to economy and reliability if there is a demand for it. The turbine designer could meet such a demand with a small power turbine which could be as reliable as the main unit which it served, and of satisfactory efficiency. As far as the author knows, however, the power-plant engineer has not been sufficiently committed to the use of the small steam turbine to warrant even tentative detail designs, although it may be practical to undertake such work in the near future. In such small-turbine design and its application to the driving of auxiliaries, it will be essential to use reduction gearing of proportions which have proved to be good for continuous service.

SPECIAL MATERIALS EMPLOYED IN THE CONSTRUCTION OF STEAM TURBINES

The use of cast steel in place of cast iron for the high-pressure sections of a turbine cylinder began with the advent several years ago of superheat temperatures of 500 deg. fahr. or more. For the cylinder or stationary parts for the highest temperatures now used, it has been found of great value to heat-treat these cast-steel parts for long 700-deg. service. The expense of annealing the cast-iron parts of the major sizes of turbines is also warranted, and adds to the assurance that the turbine will be made up of parts whose shapes and dimensions are stable. The research engineers of the company

with which the author is associated have made special investigations on the heat-treating of cast-steel parts and the annealing of cast-iron parts.[1] Extensive tests have also been made on the effect of temperature on the strength and other properties of different materials. The strengths of three materials—rolled high-chrome, high-nickel steel, forged 3½-per cent nickel steel, and forged monel metal at different temperatures—are shown graphically in Fig. 8, in which there are also indicated as curve D the properties, so far as tensile strength is concerned, which are considered desirable for a temperature of 1000 deg. fahr. There are in progress additional tests to determine the behavior of different materials when subjected to continuous stresses at constant high temperatures. These tests also cover the effect of repeated heating and cooling by steam of steam-turbine structures.

The development of satisfactory turbine structure has also

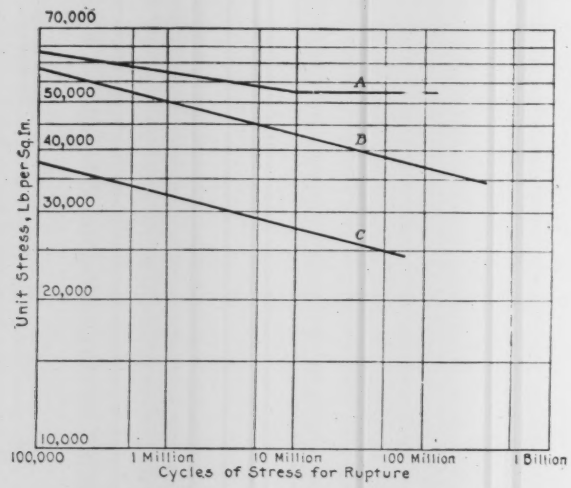

FIG. 13 EFFECT OF REPEATED STRESSES
A High-chrome, high-nickel steel
B Monel metal
C Copper-nickel

been found to depend upon the resistance of the different materials to repeated or reversed stresses, and elaborate tests have determined the resistance of the materials employed to fatigue or progressive failure. Fig. 13 shows plainly the development which has been made in this respect, the lower curve, C, being for a copper-nickel alloy which was extensively used for many years, and the upper curve, A, being for an excellent high-chrome, high-nickel steel developed for steam-turbine-blading service.

DEVELOPMENT OF NON-CORROSIVE MATERIALS

Non-corrosive metals having greater strength and resistance to corrosion than the copper and nickel alloys which have been in such long and successful use, are finding application in the steam-turbine field. In considering the application of the newer non-corrosive ferrous alloys to steam-turbine work, the author feels that there has been considerable confusion between corrosion and erosion. Non-corrosive steels have found their chief application for turbine blades and buckets and for valve-trim material. The record of machines long in service indicate that the blading material or bucket construction of a turbine is more often and most severely attacked by erosion, with corrosion as a secondary action. These remarks are to be taken as applying to machines which have not been equipped with ordinary carbon-steel blading or buckets, since all experience makes it obvious that thin-edged blades and buckets made of a steel which is subject to the slightest corrosive action would prove to have, in most cases, a relatively short life. The turbine industry from its foundation recognized the importance of non-corrosive materials for the main construction, and all departures from this fundamental principle must have been matters of expediency.

The reaction type of turbine has the erosive action on the turbine blading confined to a very few stages at the exhaust end of the machine. This condition has led to the carrying out of corrosion tests on finished turbine-blading material in service under the most severe conditions. These tests have been limited in scope because of the limited range in the reaction type of turbine in which the erosive action manifests itself on even the softest material. Corrosion, in the author's experience, has always been taken care of by the employment of materials which are not affected by the active corrosive agents which are ordinarily found in steam supplied a turbine.

The service tests mentioned have consisted of installing in the last three stages of a 3200-kw. turbine, alternate blade sections of different materials. This turbine uses a blade having a length of 6 in. for the stages subjected to the tests. The service is unusually severe since the machine operates condensing six months of the year and non-condensing the remaining six months. The results of one such series of tests are apparent from the photographs reproduced. Fig. 9 shows a section of monel-metal blading, Fig. 10 one of 3½-per cent nickel steel, Fig. 11 one of an alloy of 80 per cent copper and 20 per cent nickel, and Fig. 12 ordinary carbon steel. The materials and figure numbers are mentioned in the order of their resistance to erosion as manifested by a wearing away of the inlet blade edge directly under the shroud ring at the blade tip. The nickel steel and carbon steel, illustrated in Figs. 10 and 12 on the preceding page, were also subject to considerable corrosion, the results of which do not show plainly on the photographs, but which was sufficient to make them unsatisfactory as compared with non-corrosive material.

Cast iron maintains its tensile strength up to a comparatively high temperature, and this material should be suitable for a considerably higher temperature; but cast iron will gradually change, increasing in volume even at comparatively low temperatures over long periods of time, and is therefore not suitable except for low temperatures. It is probable that other materials which are now considered suitable for high temperatures, or new materials now being developed, will have similar characteristics when they are exposed to high temperatures over long periods of time, and it is therefore of considerable importance that this characteristic of the material be very thoroughly investigated. Undoubtedly the heat-treated materials must be tested to see if, at comparatively low temperatures, the physical characteristics will gradually change.

[1] Annealing Gray Cast Iron, by J. F. Harper and R. S. MacPherran, presented before the American Foundrymen's Association meeting, Rochester, N. Y., June 5–9, 1922. Reprinted in Allis-Chalmers Bulletin 141, Research in Materials, Nov., 1922.

Oil Resource Possibilities

A REPORT on the petroleum resources of the United States has been made to the Federal Oil Conservation Board at Washington by a committee of eleven of the directors of the American Petroleum Institute. A supply of motor fuel and lubricants sufficient to meet the needs of national defense and for essential uses beyond the time when other sources of fuel and power or less wasteful use of it shall have eliminated or restricted its use is the comforting burden of the report.

The survey of resources estimates that the petroleum recoverable from existing fields is 5,300,000 barrels, and that after pumping and flowing cease there will remain in the areas which are now producing and proved 26,000,000,000 barrels of oil, a considerable portion of which can be recovered by improved and known processes.

The major oil reserves of the United States, says the report, lie in some 1,100,000 acres of lands underlain by sedimentary rocks and not fully explored, in which geology indicates oil may be found. The nation has additional reserves in deposits of oil shale, coal, and lignites from all of which liquid fuel and lubricants may be extracted if and when the cost of recovery is justified. These deposits are so huge that they promise an almost unlimited supply.

After naming some guarantees under which the development of the industry may be fostered, the report urges the more efficient utilization of oil and improved methods of "cracking" so that the supply of gasoline will be augmented and fuel oil consequently reduced in quantity so that it will eventually be removed from direct competition with coal.

The Rational Design of Covering for Pipes Carrying Steam up to 800 Deg. Fahr.

By W. A. CARTER[1] and E. T. COPE,[2] DETROIT, MICH.

The authors present an analysis of the problem of determining the most economical thickness of pipe covering for steam pipes carrying steam of a total temperature of 500 deg. fahr. and at higher temperatures up to 800 deg. fahr. based on the work of Heilman, Eberle, and others. Two cases are considered: first, in which the total steam temperature is 500 deg. fahr. and a single covering material is used; second, in which the total steam temperature is 700 deg. fahr. and a compound covering, made up of a layer of higher-temperature-resisting material next to the pipe and a layer of lower-heat-resisting material over it. Curves showing the method of determining the most economical thickness of covering for a 6-in. pipe carrying steam at 500 deg. fahr. and 700 deg. fahr. are included. Equations for the calculations involved are included and the procedure of making the calculations is explained. Three appendices deal with some of the assumptions made in the analysis.

THE recent increase in the employment of steam having a total temperature of 700 deg. fahr. and over, makes the problem of accurately estimating and controlling the heat loss from pipes carrying superheated steam more important than ever before. Heilman[3] has produced accurate figures on the loss of heat from bare wrought-iron pipe and the loss of heat from the canvas surface of covered steam pipe. The law governing the flow of heat through thick cylindrical walls has been accurately determined.[4] But as far as the authors know, these findings have not previously been incorporated into a comprehensive method for determining the proper thickness of piping in any given case, which is the object of the present paper.

Procedure

The design of pipe covering for each size of pipe in a large central station which was recently built required the determination of the following items, estimated on the basis of 1 linear foot per year:

a Heat loss from bare pipe
b Heat loss through various thicknesses of covering
c Heat saving due to various thicknesses of covering (equals $a - b$)
d Gross monetary saving due to various thicknesses of covering (equals monetary value of heat saving in *c*)
e Fixed charges for various thicknesses of covering
f Net monetary saving due to various thicknesses of covering (equals $d - e$).

The commercial thickness of covering nearest to that showing the greatest net monetary saving was the one selected as being the most economical.

No credit was given to the pipe covering for the improvement in steam consumption of the apparatus resulting from the increase of steam temperature or steam quality, due to the reduction of heat loss from the covered pipe.

Assumptions

Cases Considered. In the first of two cases considered a single insulating material was used, while in the second case a material of higher heat-resisting properties between the pipe and the same kind of material used in the first case was considered, on account of the excessive steam temperature.

This analysis considers only a 6-in. pipe surrounded by air at 80 deg. fahr. The cost of the heat in the steam was taken as 35 cents per million B.t.u. In Case 1 the total temperature of the superheated steam was 500 deg. fahr., while in Case 2 it was 700 deg. fahr.

Temperature of Outside Surface of Pipe. The temperature on the outside surface of a bare pipe carrying superheated steam is lower than that of the steam, and the greater the amount of thermal insulation that is applied to the pipe the more closely will the temperature of the outside surface of the pipe approach that of the steam.[1] For lack of sufficient data and inasmuch as the ultimate

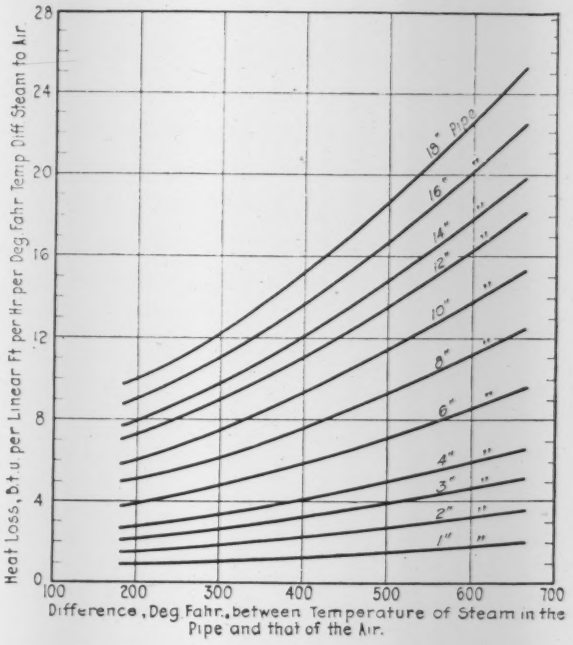

Fig. 1 Rate of Heat Transfer from Wrought-Iron Pipe to Air, at Various Temperature Differences

solution of the problem would not be influenced by the amount of temperature drop between the steam in the pipe and the outside surface of the pipe, the logic of which is set forth in an appendix to the complete paper, it was assumed that the outside surface of the bare pipe was at the same temperature as the superheated steam in the pipe.

Proof that the temperature difference between the superheated steam in an insulated pipe and the outside surface of the pipe is small, is to be found in the graphical solution of an example given in a second appendix to the complete paper, wherein the temperature drop from superheated steam to the outside of the pipe surface necessary to produce the heat flow through the covering and into the air was calculated by Eberle's formula.[2] Careful thermometric tests lead to the same conclusions.

Temperature of Inside Surface of Covering. It was also assumed that the temperature of the inside surface of the pipe covering was equal to the temperature of the pipe. There is doubtless a drop in temperature from the outside surface of the pipe to the inside

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[1] Engineer, Research Dept., The Detroit Edison Co. Assoc-Mem. A.S.M.E.

[2] Research Dept., The Detroit Edison Co.

[3] Heat Loss from Bare and Covered Wrought-Iron Pipe at Temperatures up to 800 Deg. Fahr., R. H. Heilman, Trans. A.S.M.E., vol. 44, 1922, p. 299.

[4] Mathematical Theory of Heat Conduction (Par. 31), Ingersoll & Zobel, 1913, Ginn & Co. Introduction to the Mathematical Theory of the Conduction of Heat in Solids, H. S. Carslaw (p. 114, par. 54), 1921, MacMillan Co.

Contributed by the Power Division and presented at the Spring Meeting, Milwaukee, Wis., May 18 to 21, 1925, of The American Society of Mechanical Engineers. Abridged.

[1] In the case of a pipe passing wet or dry saturated steam the temperature of the outside surface of the pipe is practically the same as that of the steam, regardless of whether it is bare or covered. This is due to the presence of a film of condensate at the same temperature as the steam on the inner surface of the pipe, and of the low resistance to heat flow from the steam through this water and the pipe wall, which results in a small temperature differential between the steam and the outside surface of the pipe for the amount of heat flowing between those two points.

[2] Mitt. über Forschungsarbeiten auf dem Geb. des Ing., Heft 78, 1909.

surface of the covering. This assumption consequently gives the greatest possible heat flow through the covering. Therefore the error, if any, is on the safe side.

HEAT LOSS FROM BARE PIPE

Heilman has reported results of experiments, conducted at the Mellon Institute of Industrial Research, to determine the rate of heat transfer from bare wrought-iron pipe of certain sizes to air at various temperature differences. Fig. 1 shows the values of the heat loss in B.t.u. per hour per linear foot of pipe per deg. fahr. temperature difference between the steam and the air, assuming that the steam and the pipe wall are at the same temperature.

HEAT LOSS FROM COVERED PIPE: CASE 1

All of the heat that escapes from the surface of the pipe covering into the surrounding air must have passed through the covering. The magnitude of this heat loss was derived from the calculated heat loss from the outside of the covering and the heat conducted through the covering for various assumed temperatures of the canvas surface. A graphical method was used to determine the heat loss that would result in a common temperature for the canvas surface.

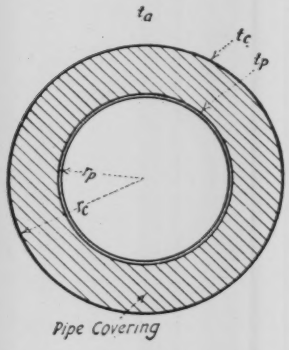

FIG. 2 SECTION THROUGH PIPE SHOWING SINGLE LAYER OF COVERING

Heilman has derived an empirical expression, based on experimental data, for the loss of heat from the canvas surface of insulated pipes. This expression is applicable to cylindrical surfaces up to 24 in. in diameter. His formula, somewhat modified, is:

$$h_1 = \frac{147.6D^{0.81}(t_c - t_a)}{272.5 - (t_c - t_a)} \quad \ldots\ldots\ldots\ldots [1]$$

in which

h_1 = B.t.u. loss per hr. per linear foot of pipe
D = diameter of the canvas surface in inches
t_c = temperature of the canvas surface in deg. fahr.
t_a = temperature of the surrounding air in deg. fahr.

The rate at which the heat passes through the layer of pipe covering conforms to the law for the passage of heat through a thick cylindrical wall. It was assumed that the pipe covering was uniform in thickness and in composition. The rate of conduction of heat through the layer of pipe covering, shown in section in Fig. 2, was expressed by the equation:[1]

$$h_2 = \frac{0.524K(t_p - t_c)}{\log_e \frac{r_c}{r_p}} \quad \ldots\ldots\ldots\ldots [2]$$

in which

h_2 = B.t.u. per hr. per linear foot of pipe flowing from inner surface to outer surface of pipe covering
K = mean absolute thermal conductivity of the insulating material expressed in B.t.u. per sq. ft. per hr. per in. of thickness per deg. fahr. temperature difference between the hot and cold surfaces. K is not a constant but changes with the mean temperature of the material
t_p = temperature of the inside surface of the layer of pipe covering in deg. fahr. This is taken the same as the temperature of the steam
t_c = temperature of the outside surface of the layer of pipe covering in deg. fahr.
r_c = radius, in inches, of the outside surface of the pipe covering
r_p = radius, in inches, of the inside surface of the pipe covering. This was taken equal to the outside radius of the pipe.

Assuming that thermal equilibrium has been established, the heat loss from the canvas covering per linear foot of pipe must equal the heat conducted through the covering per linear foot of pipe. Consequently, Equations [1] and [2] become equal to each other; that is,

$$\text{Heat loss} = \begin{bmatrix}\text{Heat loss from} \\ \text{canvas surface}\end{bmatrix} = \begin{bmatrix}\text{Heat conducted through the} \\ \text{covering}\end{bmatrix}$$
$$= h_1 = h_2$$

A graphical method was used to solve Equations [1] and [2] simultaneously. In Equation [1] the known quantities were t_a and D, which latter equals 2 × (r_p plus assumed thickness of covering). Values of h_1 were calculated for assumed values of t_c, namely, 100, 150, and 200 deg. fahr. The curve marked "Loss from Covering to Air," Fig. 3, is a plot of the results.[1]

In Equation [2] the known quantities were K taken from a curve for the average temperatures of the covering as fixed by t_p and the three assumed values of t_c, t_p, r_p, and r_c, which equals r_p plus assumed thickness of covering. Again, various values for t_c were assumed and the equation solved for h_2. These values were plotted on the same sheet as the loss-from-covering-to-air curve, Fig. 3. The values for t_c were taken as 70, 200, and 300 deg. fahr. This curve is marked "Heat Passing through Covering."

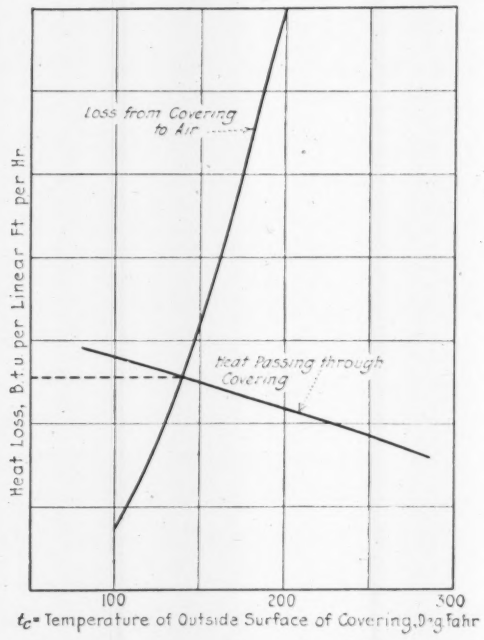
Loss from Covering to Air
Heat Passing through Covering
Heat Loss, B.t.u. per Linear Ft. per Hr.
100
200
300
t_c = Temperature of Outside Surface of Covering, Deg. Fahr.

FIG. 3 HEAT LOSS THROUGH 1½-IN. COVERING ON A 6-IN. PIPE

The point of intersection of these two curves indicates a condition of thermal equilibrium for the assumed thickness of covering. It is then possible to pick off the loss in B.t.u. per linear foot of pipe by taking the value of the ordinate of this intersection. The abscissa of this point of intersection indicates the actual temperature of the outside of the covering.

The B.t.u. loss per linear foot of pipe was also obtained for several thicknesses of covering. By plotting these values of B.t.u. loss per linear foot of pipe having different thicknesses of covering against the thickness, Fig. 4 was obtained. The heat loss per linear foot of the bare pipe as obtained from Fig. 1 also was plotted in Fig. 4. This curve shows the loss of heat from the 6-in. pipe carrying steam at 500 deg. fahr., both bare and when protected with the particular covering considered in thicknesses up to 5 in.

HEAT SAVING DUE TO COVERING

The difference between the bare-pipe loss and the loss through any particular thickness of the pipe covering is the saving in B.t.u.

[1] Introduction to the Mathematical Theory of the Conduction of Heat in Solids, H. S. Carslaw (p. 114, par. 54), 1921, MacMillan Co. Mathematical Theory of Heat Conduction, Ingersoll & Zobel (par. 31), 1913, Ginn & Co. Equation [2] has been modified to give h_2 in B.t.u. per hr. per linear foot of pipe instead of in B.t.u. per hr. per sq. ft. of cylindrical surface.

[1] The actual values of heat loss in this and subsequent figures are omitted, as they are not essential to demonstrate this method of solution.

per linear foot of pipe per hour. This saving is effected by the presence of that thickness of the pipe covering. Fig. 5 shows the heat-saving curve for a 6-in. pipe carrying steam at 500 deg. fahr. The ordinates of this curve were obtained from Fig. 4. The curve shows several striking characteristics. It shows in particular that the first quarter of an inch thickness of pipe covering is of more value in saving heat than any additional inch thickness of the same material.

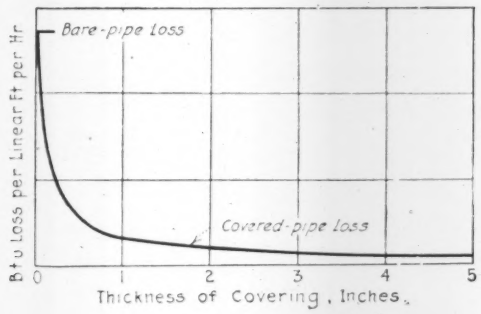

FIG. 4 B.T.U. LOSS FROM A 6-IN. PIPE CARRYING STEAM AT 500 DEG. FAHR.

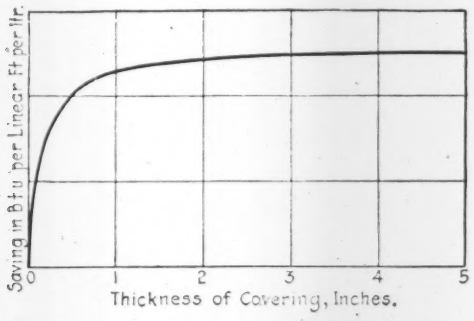

FIG. 5 B.T.U. SAVING RESULTING FROM VARIOUS THICKNESSES OF PIPE COVERING ON A 6-IN. PIPE CARRYING STEAM AT 500 DEG. FAHR.

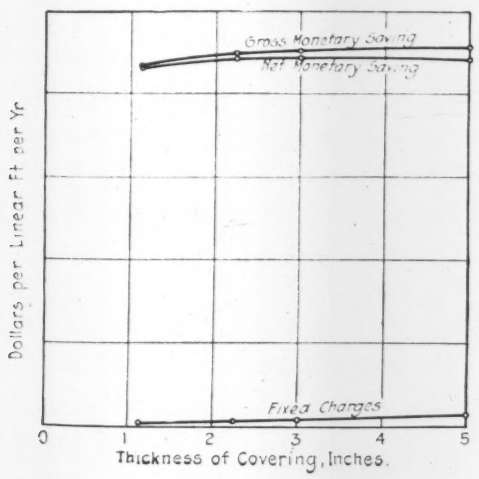

FIG. 6 CURVES USED IN SELECTING THE MOST ECONOMICAL THICKNESS OF COVERING FOR A 6-IN. PIPE CARRYING STEAM AT 500 DEG. FAHR.

GROSS MONETARY SAVING DUE TO COVERING

The gross monetary saving is the monetary value of the heat saving. In this study the cost of steam, taken from the operating records of a certain large plant for a recent annual period, was 35 cents per million B.t.u. in the steam. Any other reasonable cost would have been equally satisfactory. The gross-monetary-saving curve for the case under consideration is shown in Fig. 6. The effect of a different cost per million B.t.u. upon the most economical thickness is discussed in a third appendix to the complete paper.

FIXED CHARGES

The fixed charges on the covering, including maintenance, depreciation, interest, and taxes, were taken as 20 per cent of the cost of the covering in place. The cost per linear foot of covering of any particular thickness includes the cost of the materials, freight charges, and the cost of labor to place the covering on the pipe. The cost of labor for installing the covering was an average estimating cost obtained from two nationally known contracting companies. The fixed-charges curve was plotted as shown in Fig. 6. As the thickness of covering increased, the slope of this curve increased.

NET MONETARY SAVING DUE TO COVERING

The net-monetary-saving curve shown in Fig. 6 is the difference between the gross-monetary-saving curve and the fixed-charges curve. The net-monetary-saving curve contains the answer of this problem. The flatness of the curve indicates that there is a rather wide latitude in the choice of the proper thickness of covering. Fairly wide variation in thickness causes little difference in net monetary saving.

Steam-pipe covering is supplied in definite commercial thicknesses. Some are made up of one layer and others of two layers. It is therefore possible to select the commercial thickness which most nearly fits the conclusions shown in Fig. 6. A thickness of 3 in. seemed to be about the best for the case considered. A covering made up of two layers of $1^1/_2$ in. thickness each was the one chosen for the particular material considered.

HEAT LOSS FROM PIPES HAVING COMPOUND COVERING: CASE 2

In the case just discussed a single covering of 85 per cent magnesia was considered on pipes carrying steam up to 500 deg. fahr. For pipes carrying steam at temperatures in excess of 500 deg. fahr., the covering was made up of a protective layer next to the pipe and 85 per cent magnesia over this protective layer. Investigations have disclosed that 85 per cent magnesia insulation undergoes a change when heated continuously at a temperature ranging from 500 to 600 deg. fahr., which change weakens the material mechanically. In consequence of these observations, it was decided arbitrarily to set 500 deg. fahr. as a maximum temperature to which the inside of the layer of 85 per cent magnesia covering would be subjected. A material which would not show serious loss of mechanical strength at a temperature higher than 500 deg. fahr. was selected to be used next to the pipe.

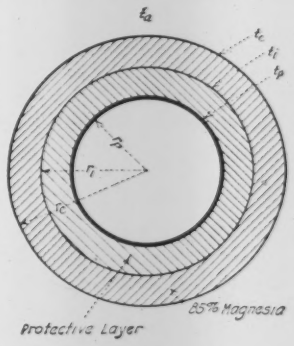

FIG. 7 SECTION THROUGH PIPE SHOWING COMPOUND COVERING

When such a combination of pipe covering, made of layers of two different materials as shown in Fig. 7, was used, the design method was changed only in that the two materials would not ordinarily have the same thermal-conductivity values. Equation [3] for the thermal equilibrium was expanded to include the passage of heat through a second layer of insulating material. This equation for compound covering became

$$\text{Heat loss} = \begin{bmatrix}\text{Heat loss}\\ \text{from can-}\\ \text{vas cover}\end{bmatrix} = \begin{bmatrix}\text{Heat conducted}\\ \text{through outer}\\ \text{layer}\end{bmatrix} = \begin{bmatrix}\text{Heat conducted}\\ \text{through pro-}\\ \text{tective layer}\end{bmatrix}$$

$$= h_1 = h_3 = h_4$$

$$h_1 = \frac{147.6D^{0.81}(t_c - t_a)}{272.5 - (t_c - t_a)} \quad \ldots\ldots [1]$$

$$h_3 = \frac{0.524K_a(t_i - t_c)}{\log_e \frac{r_c}{r_i}} \quad \ldots\ldots [3]$$

$$h_4 = \frac{0.524K_b(t_p - t_i)}{\log_e \frac{r_i}{r_p}} \quad \ldots\ldots [4]$$

The symbols are identical with those used in Case 1, with the addition of K_a, K_b, r_i, and t_i.

h_3 = B.t.u. conducted through the outer layer per hour per linear foot of pipe

h_4 = B.t.u. conducted through the protective layer per hour per linear foot of pipe

K_a = mean absolute thermal conductivity of the outside layer. K_a is expressed in B.t.u. per sq. ft. per hr. per in. thickness per deg. fahr. temperature difference between the inside and the outside of this layer of material

K_b = a similar value to K_a for the protective layer of the compound covering

r_i = radius in inches of outside surface layer of protective material

t_i = temperature of outside of layer of protective material. It was assumed that there was no drop in temperature between the outside surface of the protective layer and the inside surface of the outside layer.

The solution of this set of equations was carried out as follows:

The loss of heat from the canvas surface was calculated from Equation [1]. Values of t_c were assumed after D had been taken at some convenient value. The total covering thickness on a 6-in. pipe was taken equal to 4 in. Therefore D equaled the outside diameter of the pipe plus 8 in.; t_c was assumed as 80, 100, and 150 deg. fahr., and the equation was solved for each value of t_c. Values of h_1 were thus obtained for these values of t_c. The loss-from-canvas-surface curve was plotted between h_2 and t_c. This is shown on the right-hand side of Fig. 8 and is marked "Loss from Canvas Surface."

Next, the heat passing through the outside layer of the covering was computed by the use of Equation [3]. In this equation K_a was obtained from the thermal-conductivity curve of the material for the temperature $\frac{(t_i + t_c)}{2}$. The temperature of the inner surface of the outer layer of covering, t_i, was taken as 500 deg. fahr., the assumed permissible maximum temperature. The quantity r_i was taken at several convenient thicknesses of the inside layer of the covering. Values of r_i equal to 3.3125, 4.3125, and 5.3125 in. were obtained by assuming thicknesses of protective covering equal to 0, 1, and 2 in., respectively. Values of t_c of 100, 200, and 300 deg. fahr. were assumed for each value of r_i. The various values of h_3 were calculated and a curve plotted between h_3 and t_c for each value of r_i. This gave the family of curves marked "Loss through Outer Covering," Fig. 8. These curves intersected the loss-from-canvas-surface curves at values of heat loss and t_c representing thermal-equilibrium conditions.

Next, a curve plotting the ordinates of these points of intersection as ordinates and the corresponding radii (r_i) as abscissas was laid out on the left-hand side of the sheet. This is curve Y, Fig. 8.

Then, values of heat passing through the layer of protective material were calculated by means of Equation [4]. In this equation, r_i was the only unknown since t_p was assumed equal to the temperature of the steam and t_i equaled 500 deg. fahr., by assumption. Substituting in Equation [4] the same values of r_i used in Equation [3], the various values of h_6 were calculated. Then the curve which is marked "Loss through Protective Covering" was plotted on the left-hand side of Fig. 8 between the values of h_6 just calculated as ordinates and the corresponding radii (r_i) as abscissas.

Thermal equilibrium of the system between the steam in the pipe and the surrounding air occurs, for the conditions assumed, when the loss in B.t.u. per linear foot of pipe per hour equals that represented by the ordinate of the intersection of curve Y and the loss-through-protective-covering curve. If a horizontal line be drawn toward the right from this point of intersection until it intersects the curve marked "Loss from Canvas Surface," the temperature of the outside surface of covering corresponding to this point is the temperature of the outside surface of the covering for thermal-equilibrium conditions. The ordinate of this point of intersection is the heat loss per linear foot of pipe under the same conditions.

At the top of Fig. 8, toward the left, are two scales, one marked "Thickness of Protective Covering," the other marked "Thickness of Outer Covering." The total covering thickness used in this solution was 4 in. This may be made up of 1 in. of protective layer and 3 in. of outer layer, or 2 in. of each, or 3 in. of protective layer and 1 in. of outer layer. A 1-in. protective layer makes an outside radius of protective covering on a 6-in. pipe equal to 4.3125 in. On the thickness-of-protective-layer scale at the top of Fig. 8, 1, 2, and 3 correspond with 4.3125, 5.3125, and 6.3125, respectively, on the outside-radius-of-protective-covering scale at the bottom of Fig. 8. A vertical line drawn upward from the intersection of curve Y and the loss-through-protective-covering curve intersects the thickness-of-protective-covering scale at $1^3/_{16}$ in. The thickness of the outer layer is $2^{13}/_{16}$ in. (equals $4 - 1^3/_{16}$). These dimensions satisfy the condition imposed by the effect of high temperature on the 85 per cent magnesia, namely, that the temperature of the inside surface of the outer layer shall be 500 deg. fahr.

All that it was necessary to do then was to plot three or four of these curves for different total thicknesses of covering, take off the loss of heat in B.t.u. per linear foot of pipe and plot the heat-saving curves for various total covering thicknesses as shown in Fig. 5. From this point the procedure was the same as is described in the early part of this paper. When the analysis was completed it was necessary to select the thicknesses for both the protective layer and the outside layer that were obtainable in commercial sizes of pipe coverings.

If the thickness of the protective layer obtained was not a commercial standard, a greater rather than a lesser thickness than that dictated by the analysis was chosen, thereby subjecting the 85 per cent magnesia to a temperature lower than the maximum arbitrarily set.

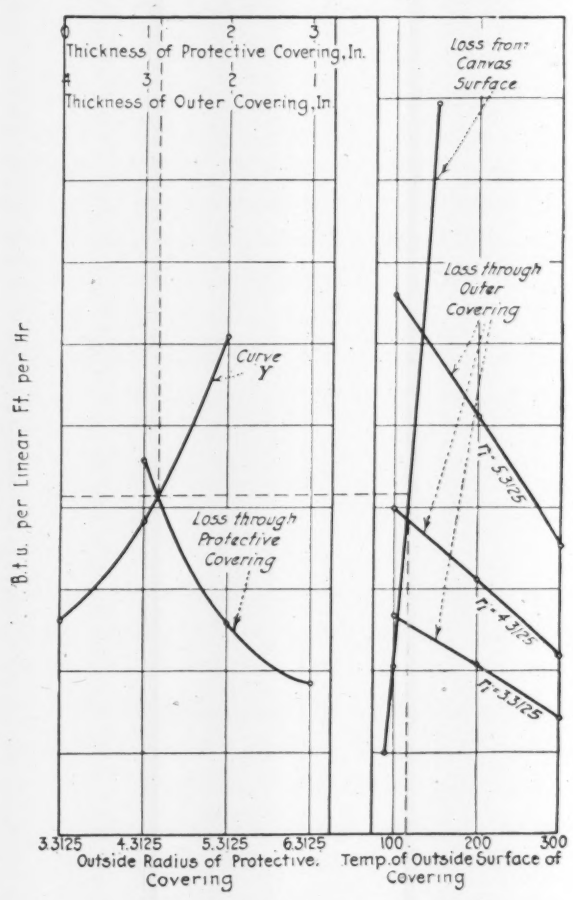
0 1 2 3
Thickness of Protective Covering, In.
3 2 1
Thickness of Outer Covering, In.
Loss from Canvas Surface
Loss through Outer Covering
Curve Y
Loss through Protective Covering
r_i = 5.3125
r_i = 4.3125
r_i = 3.3125
B.t.u. per Linear Ft. per Hr.
3.3125 4.3125 5.3125 6.3125
Outside Radius of Protective Covering
100 200 300
Temp. of Outside Surface of Covering

FIG. 8 DETERMINATION OF HEAT LOSS PER LINEAR FOOT FROM A 6-IN. PIPE HAVING 4 IN. OF COMPOUND COVERING AND CARRYING STEAM AT 700 DEG. FAHR.

It is obviously out of place in such a paper as this to go into a discussion of actual materials that lend themselves readily for use in the protective layer. It is equally obvious that exact values of thicknesses of pipe covering cannot be stated because different brands of these materials have different thermal conductivities and are sold at different prices.

Lake Waters for Condensers

Persistence of Stratification of Warm- and Cold-Water Layers in Northern Lakes—Savings from Increased Vacuum Obtained by Use of Cold Bottom Layers in Condensers

By A. G. CHRISTIE,[1] BALTIMORE, MD.

WHEN large power stations are located on lakes or other large bodies of quiescent water in the northern section of this country, there are certain phenomena connected with the stratification of such waters that should be given careful consideration in the design of cooling-water intakes. This paper will discuss these phenomena and will indicate the large economic savings that may be realized by the proper design of intakes to take full advantage of the cold waters that are found in the lower strata of these lakes.

Factors Affecting Temperature Variation in Lakes

The northern lakes are generally frozen over completely or have surface waters which drop to nearly 32 deg. fahr. for many months each winter. The surface if frozen will be at 32 deg. fahr. The water at the bottom of the lake is generally 34 to 37 deg. fahr. during the winter; and little mixing takes place since the water at this temperature is more dense than at 32 deg. fahr.

After the ice has left the lake, the surface waters warm up gradually to 39.2 deg. fahr., the lower water remaining practically unchanged in temperature. Water has its greatest density at 39.2 deg. fahr., and if the bottom water is at a lower temperature, say, at 35 deg. fahr., it will rise to the surface to be displaced by the dense surface water at 39.2 deg. fahr. This results in a complete displacement of the water at the bottom by surface water. Thus the water of these lakes is overturned twice a year, in spring and again in the fall. This overturn is brought about partly by the wind and partly by convection currents due to differences in density. The wind is an important factor in bringing about this overturn and particularly in continuing it. The dense surface water is often blown to the leeward side of the lake where it sinks to the bottom as a means of returning to maintain the water level on the windward side. This displaces the bottom water and starts currents which rapidly overturn the whole lake. The mixing process continues until practically all of the water is mixed up and the whole of the lake approaches a constant temperature.

The circulation often continues for a month or more in many lakes, for although the surface waters heat up in day time, they cool off at night and the convection currents are started again. This circulation frequently continues until the whole lake has increased to 45 deg. fahr. in temperature. However, at some time in the late spring the sun will warm the surface waters so that their temperature will not fall during the night below that of the lower layers, and as soon as this happens, circulation ceases. The lower waters never rise again to the surface of the lake during the whole summer, and, as will be shown later, their temperature throughout that period rises only a few degrees above that at which circulation stopped. This is a very important fact to power engineers.

The value of the thermal resistance to mixture is of interest for it shows why the winds can no longer cause mixing of the waters at various depths. The thermal resistance depends solely on the change of density of the water as its temperature rises above 39.2 deg. fahr. At first the change in density for each degree rise is small, but this change per degree increases rapidly with rise in temperature above 39.2 deg. fahr. The following example will serve to show the increasing rate of change of density: A given volume of water weighing 1 lb. at 49 deg. fahr. weighs 0.000045 lb. less at 50 deg. fahr. for the same volume; another volume weighing 1 lb. at 76 deg. fahr. weighs 0.000142 lb. less at 77 deg. fahr. for the given volume. The rate of change in density per degree at 77 deg. fahr. is thus over three times the rate of change at 50 deg. fahr., and hence the thermal resistance per degree is increased.

Studies of density changes indicate that the thermal resistance to mixing is small for a few degrees either side of 39.2 deg. fahr., so that the lake waters frequently reach temperatures of 40 to 45 deg. fahr. before the spring overturn ceases, particularly if strong winds prevail. However, increasing surface temperatures soon increase the rate of change of density so rapidly that this mixing must cease. The wind effects do not penetrate to the lower depths. Convection currents due to differences of density have ceased and water is a poor conductor of heat. Hence the layers near lake bottoms receive no further heat and remain undisturbed at practically the same temperature throughout the whole summer and well into the fall.

Observations taken upon lakes throughout the summer have indicated the same temperature as at the surface for the first few feet and often for 10 or 12 ft. in depth. This would indicate that the winds keep the water stirred up more or less throughout this depth. Below 12 ft. the temperature decreases until at a lower depth, depending on the particular lake, the cold strata are reached where the temperature is constant for the whole summer. The coldest waters are encountered at different depths, depending on the deepness of the lake. This depth varies from 25 to 30 ft. from the surface for lakes about 65 feet deep, while at Lake Thun in Switzerland, with a total depth of 550 ft., the cold layers start at about 125 ft.

The discontinuity layers of varying temperature that lie below the warm surface water have been called the "thermocline." The fall in temperature at the junction of the thermocline and the warm surface layers is often very rapid. Observations have frequently indicated a drop at this junction of 7 to 9 deg. fahr. in three feet of depth when the surface temperatures are above 60 deg. fahr. In some cases there even appears to be a sharp dividing line between the surface water and the thermocline. Such a drop in temperature offers a very great thermal resistance to mixture by convection. There is a much less abrupt change in temperature between the layers of cold water at the bottom and those immediately above them as is indicated by accompanying curve and table. The position of the thermocline when once established is altered little throughout the summer, even by violent and long-continued winds.

TABLE 1 VARIATION OF TEMPERATURE WITH DEPTH, GREEN LAKE, WIS.

Depth in feet	Temperatures, deg. fahr.								
	Aug. 20, 1905	Feb. 15, 1906	May 15, 1906	July 5, 1906	Aug. 14, 1906	Oct. 4, 1906	Sept. 14, 1907	Sept. 3, 1908	Aug. 20, 1909
0.0	73.4	32.0	...	67.6	76.1	64.2	65.7	68.7	78.1
32.8	69.2	33.5	...	66.2	68.2	63.9	63.5	68.3	70.7
39.4	63.1	...	...	56.8	61.7	63.9	57.4	64.4	60.1
49.2	54.7	...	...	50.2	54.1	54.1	49.5	52.3	53.2
65.6	47.5	33.5	42.1	45.1	46.8	46.6	45.8	45.8	46.4
98.4	45.3	34.0	42.1	43.9	44.2	44.0	43.9	44.2	44.8
131.2	43.1	34.7	41.2	43.3	44.2	43.2	42.4	42.6	44.4
152.4	42.8	35.0	40.6	42.8	43.0	42.6	42.1	42.4	44.1
182.9	42.4	35.4	40.5	42.3	42.4	...	41.7	42.3	43.7
213.4	42.3	36.1	40.5	41.5	42.3	42.3	41.4	41.9	43.7

Table 1, from The Inland Lakes of Wisconsin,[1] shows the variation of temperature with depth in Green Lake. This is the deepest inland lake in Wisconsin. Its area is about 11 square miles. Sixty per cent of this area is 65 ft. or more in depth, while 50 per cent is over 100 ft. in depth. The lake bottom rises steeply within a few hundred feet of some of its shores so that its lower waters would be easily accessible for power-plant use. An interesting point indicated by this table is that the temperatures of the lower strata, while constant throughout each season, may vary slightly in different years, depending on the temperature at which the spring overturn ceased.

Fig. 1, from The Temperature of Lakes,[2] by Desmond Fitzgerald, shows graphically the variation of temperature with depth

[1] Professor of Mechanical Engineering, Johns Hopkins University. Mem. A.S.M.E.

Contributed by the Power Division and presented at the Spring Meeting, Milwaukee, Wis., May 18 to 21, 1925, of The American Society of Mechanical Engineers. Abridged.

[1] Bulletin 22, Scientific Series No. 7, Wisconsin Geological and Natural History Survey, 1911.

[2] Trans. A.S.C.E., 1895.

for Lake Cochituate near Boston. The undisturbed cold lower layers are apparent. This paper also contains some observations made on Lake Superior which show that with surface temperatures of 50 to 56 deg. fahr. the deep-water temperatures range from 36.8 to 39 deg. fahr.

Certain lakes in Scotland and Switzerland have been studied and show the same temperature stratification. Shallow lakes or other shallow bodies of water not over 25 ft. in depth are not likely to exhibit this phenomenon for they are too much stirred up by the winds.

TEMPERATURE VARIATIONS IN BAYS, SOUNDS AND OCEAN

Through the courtesy of Dr. R. P. Cowles of Johns Hopkins University, certain temperature-depth observations of the Hydrographic Survey, Bureau of Fisheries, on Chesapeake Bay were made available for study. Unlike the lakes just described, there does not seem to be any definite thermocline in the bay nor any cold bottom layers. There is some variation of temperature with depth but it

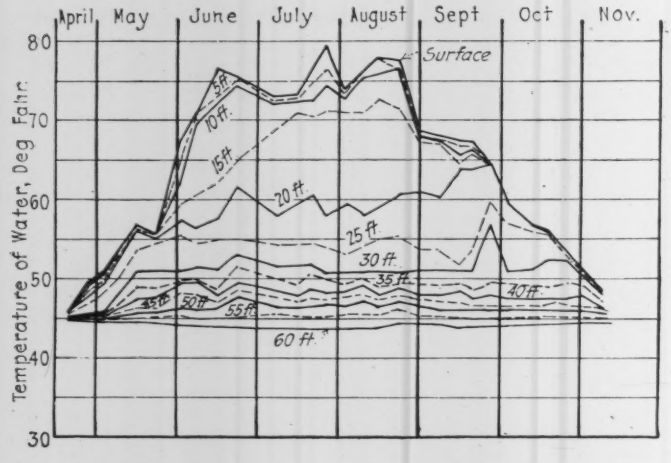

FIG. 1 TEMPERATURE VARIATIONS WITH DEPTH IN LAKE COCHITUATE

is not large. Though large in area and with deep channels, the average depth of the bay is not very great. It has strong currents from tides and receives large discharges from fresh-water rivers. The winds stir it up easily in summer and mix its waters. Thus every agency works against the establishment of a distinct stratification of its waters.

Observations made outside Cape Henry in the Atlantic Ocean indicated quite a different state of affairs. With surface temperatures of 70 to 75 deg. fahr. the temperatures at 60 ft. depth averaged about 42 deg. fahr., and this persisted quite close into shore. One can therefore expect to find stratification along the sounds and bays of our coast where reasonable depths occur and where tides and currents do not disturb the waters too much.

APPLICATION TO CONDENSERS

The intakes for cooling water at power plants located on such bodies of water as above described, would probably lie from 15 to 20 ft. below the normal surface. Such intakes will draw their water from the thermocline where the temperature varies during the season, as shown by Fig. 1. A study of the data here presented and of other available data indicates that water 10 to 20 deg. fahr. colder could be obtained by intakes that would reach the cold bottom layers. Power stations on such lakes usually carry vacua ranging from 28 to 28$^1/_2$ in. during the summer months. An inch of vacuum can be allowed for every 22 deg. fahr. change in intake temperature. With deep-water intakes and colder water it seems probable that 29 in. vacuum or higher could be maintained throughout the whole year. The possibility of such a condition is illustrated by Fig. 2 which is plotted from the observations in Table 1 for Feb. 15 and Aug. 14, 1906. The location of a condenser intake that would draw water from 75 ft. depth instead of 20 ft. depth would provide on the August day cooling water at about 45 deg. fahr. instead of 73 deg. fahr. This would develop an inch and a quarter more vacuum in the condenser. Furthermore with the deep intake there would only be a possible variation of cooling-water temperature from about 34 deg. fahr. in winter to 45 deg. fahr. in summer, which would permit the design of a plant for at least a 29-in. vacuum throughout the year.

The decrease in steam consumption of a large turbine for a change in vacuum from 28 to 29 in. is about 7$^1/_2$ per cent, and from 28$^1/_2$ to 29 in. about 4 per cent. The value of additional vacuum due to colder cooling water may be indicated by the following figures, assuming that the average vacuum was increased only from 28$^1/_2$ to 29 in. for 4$^1/_2$ months of the summer. Take a plant capacity of 100,000 kw.; a use factor over this period of 40 per cent; fuel consumption of 1$^1/_2$ lb. of coal per kw-hr.; coal cost, $5 per ton of 2000 lb.; and 4 per cent decrease in steam consumption due to increase of vacuum from 28$^1/_2$ to 29 in.

Savings for 4$^1/_2$ months

$$= \frac{4.5 \times 8760 \times 100{,}000 \times 0.40 \times 1.5 \times 0.04}{12 \times 2000}$$

$$= 3942 \text{ tons of coal valued at } \$19{,}710$$

If the vacuum had been increased from 28 to 29 in. with 7$^1/_2$ per cent less steam consumption the saving would have amounted to $36,956. No allowance has been made in these figures for the lower cost of pumping as a result of the probable use of smaller quantities of colder water at light loads.

The rate at which these savings of $19,710 and $36,956 per annum could be capitalized depends upon cost of money, taxes, and other conditions, among which are the number of years of service to be secured from such extra length and depth of tunnels and intakes, and maintenance and possible alterations in the meantime. If all these conditions are such that the foregoing savings could be capitalized at 12 per cent, then approximate additional

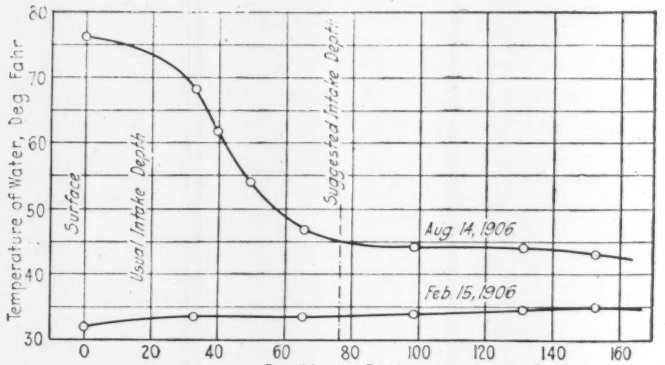

FIG. 2 RELATION BETWEEN TEMPERATURE AND DEPTH ON TYPICAL MIDWINTER AND MIDSUMMER DAYS IN GREEN LAKE, WIS.

expenditures of $164,000 and $308,000, respectively, would be justified for a plant of 100,000 kw.

One will naturally say that these lower depths are not accessible to most plants as the shore waters are shallow. This is frequently the case, and open cuts to deep water are objectionable for they silt up quickly. Under certain conditions the savings from the colder water may warrant the construction of a water-works type of tunnel with a deep-water intake. If the distance is not great, a steel or concrete pipe might be laid on the lake bottom to the deep water. The additional pumping head will only be that due to friction in the additional length of conduit, and this may be offset by careful design of the deep-water intake so as to eliminate most of the losses that now occur in an open-ended intake. The discharge tunnel for the cooling water from the condensers should spread out near the surface in a wide, shallow mouth with low velocity so as to distribute the hot water only to the surface layers of the lake and not force it into the colder lower layers.

There is little danger of using up the cold bottom layers during the season, and with a proper design of intake it should be possible to take in only the colder layers. During the summer period there is a tendency for the organic matter from the upper layers to collect in the lower layers, and this organic matter would pass into the condenser. Organic material gives little trouble from slimes and growth in condenser tubes with cold condensing water, and hence would be of no consequence on account of the low temperatures.

Comparison of Actual Performance and Theoretical Possibilities of the Lakeside Station

By M. K. DREWRY,[1] MILWAUKEE, WIS.

The object of this paper is to show the degree of perfection in the design and operation of a high-efficiency, straight-Rankine-cycle station, pointing out the magnitude of each individual heat and availability loss occurring in actual practice. Operating and test figures of the Lakeside station have been assembled, and a comparison of theory and practice has been recorded.

TO PREDICT accurately the efficiency of a power station is more involved than to ascertain the economy of the theoretical cycle it employs. To record and explain the extent and cause of the difference between theory and practice in one particular example, that of the Lakeside station, is the primary purpose of this paper.

Comparison of theoretical possibilities and actual performance has been made during a given period of best economy, October 10 to November 10, 1924, by the following means:

1 Computing plant "efficiency ratio" when the station is most economically loaded, estimating 2 per cent radiation and miscellaneous losses

2 Finding losses attendant with actual operation due to uneconomical loading of turbines and boilers and to starting and banking, etc., using actual operating data

3 Calculating the theoretical cycle efficiency with which a perfect station would have worked, and multiplying this cycle efficiency by the efficiency ratio of the actual station

4 Equating actual and calculated heat consumptions to check allowances made in the calculations for radiation and miscellaneous losses.

The calculations of Table 1 show that less than 5 per cent (see item 10) need be added to the best theoretical performance of the station to obtain its heat consumption under operating conditions of maximum economy. In terms recently introduced, its "operating efficiency ratio" is approximately 95 per cent.

Of the 5 per cent difference, about one-quarter of one per cent can be attributed to uneconomical boiler loading, 2 per cent to banking and starting cold boilers, 2 per cent to uneconomical loading and starting of turbines, and the remainder to unusual operations and miscellaneous losses. The theoretical-plant-efficiency calculation allows 2.2 per cent for radiation and miscellaneous losses, a large share of which are constant regardless of load carried.

TABLE 1 COMPARISON OF THEORETICAL POSSIBILITIES AND ACTUAL PERFORMANCE

	Per cent
Most Economical Loading	
1 Boiler efficiency (computed from losses)	91.6
2 Turbo-generator efficiency ratio (from tests)	70.2
3 Consumption of auxiliaries	
Electrical auxiliaries (from meters)	4.25
Steam (credit given for heat return to condensate; taken from operating data)	1.84
4 Coal drying (from operating data)	0.25
5 Radiation and miscellaneous losses (by difference to equalize items 14 and 15)	2.2
6 Efficiency ratio,[1] most economical loading	64.0
Losses Attendant with Actual Operation (Calculated from Operating Data)	
7 Boiler room	
Uneconomical loading of boilers	0.25
Banking and starting cold boilers	1.98
8 Turbine room	
Uneconomical loading	1.75
Starting	0.26
9 Unusual operations	
Entire plant	0.22
10 Total operating losses	4.49
11 Calculated efficiency ratio, average operating conditions	61.1
12 Theoretical cycle efficiency (Marks and Davis tables)	35.4
13 Calculated plant thermal efficiency	21.6
	B.t.u. per kw-hr.
14 Corresponding calculated heat consumption	15,800
15 Actual plant heat consumption	15,800

[1] A measure of equipment efficiency. Actual boilers, turbines, and auxiliaries have a combined efficiency of 64 per cent that of perfect equipment.

TABLE 2 HEAT BALANCE

	B.t.u.	Per cent
Heat per lb. of coal	13,691	100
Heat loss due to moisture in coal	26	0.19
Heat loss due to water from combustion of hydrogen	487	3.55
Heat loss due to moisture in air	10	0.07
Heat loss in dry chimney gases	453	3.31
Heat loss due to incomplete combustion of carbon	68	0.5
Heat loss due to unconsumed hydrogen, hydrocarbons	0	0
Heat radiated but not reclaimed; and unaccounted-for losses	103	0.75
Heat absorbed by boiler, economizer, and superheater	12,544	91.6

Basis of Calculations

Boiler Efficiency. Continuous measurement of the most significant measurable losses in boiler operation allows boiler efficiencies to be quite accurately deduced by totalizing these comparatively small losses instead of equating the large inputs and outputs as done in complete boiler testing. Table 2 is a heat balance calculated from data gathered during the period in question.

No attempts to learn the percentage of combustible in the flue gases have as yet been successful, for the particles of ash have been found so extremely fine that no convenient filter will remove proportional quantities of ash and coal. The solids emerging from the stacks during normal operation appear no darker than pure calcined ash, but inability to measure the carbon losses during periods of starting, when they are of some consequence, has required an estimate of 0.5 per cent to be used.

Loss due to unconsumed hydrocarbons cannot be detected in the furnaces until excess air is decreased below 15 per cent, and since the average operating value of excess air is 22 per cent, the loss is negligible.

Radiation losses are undoubtedly small. Furnace-casing temperatures average about 110 deg. fahr., and practically all heat losses contribute, with the supply of heat in the generator cooling air, to maintain the combustion-air temperature at 100 deg. fahr. The coolest daily average combustion-air temperature recorded the past winter was 80 deg. fahr. Previous tests on units in the first boiler installation placed radiation and unaccounted-for losses at 2 per cent, but the newer boilers, which carried the largest percentage of the load during the period considered, have considerably lower radiation losses.

Turbo-Generator Efficiency. This is a weighted value derived from test results averaging the Rankine-cycle efficiency ratio (Marks and Davis steam tables) of the four units installed and operated during the time considered. The efficiency ratio recorded includes generator losses and excitation energy.

Consumption of Auxiliaries. Credit for heat returned to the condensate by exhaust steam from the auxiliaries has been allowed. Table 3 shows the itemized auxiliary consumptions.

Drying of Coal. The boiler efficiency recorded in the previous tabulations does not consider coal-drying charges, though the usual monthly efficiencies tabulated elsewhere do.

The average summer and winter values of the coal used per ton dried per 1 per cent of moisture removed, are about 2 and 5 lb., respectively. Table 4 gives the average operating figures for the period considered.

Radiation and Miscellaneous Losses. The value of 2.2 per cent gave agreement between calculated and actual results.

Chief of the unreclaimed radiation losses is the 25-deg. drop in superheated-steam temperature from boiler room to turbine room. Blow-down losses are slight, 6 B.t.u. per kw-hr., the character of Lake Michigan water being such that the few dissolved solids settle out at the blow-off pads, and that the concentration of the boiler water is caused to remain continuously very low, less than 0.1 deg. B.

Another source of loss is that of steam leakage through radiant-superheater elements. The period of best economy chosen for this analysis occurred at a time when station make-up was lowest,

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Contributed by the Power Division and presented at the Spring Meeting, Milwaukee, Wis., May 18 to 21, 1925, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. Abridged.

TABLE 3 DIVISION OF ENERGY TO AUXILIARIES

	Per cent
A—Electrical Auxiliaries	
1 Station lighting	0.1
2 Coal handling:	
Car dumper	0.0006
Magnetic pulley	0.006
Crushers	0.02
3 Preparation house:	
Conveying green coal	0.003
Pulverizing	1.08
Conveying pulverized coal	0.025
Total	1.110
4 Boiler room: [1]	
Induced-draft fans	0.50
Coal-feeder motors	0.08
Primary air fans	0.25
Boiler-feed pumps	0.62
Air compressor, machine shop, etc.	0.25
Total	1.65
5 Turbine room:	
Circulating pumps	0.88
Hotwell pumps	0.29
Air removal (on one unit)	0.17
House service and miscellaneous	0.03
Total	1.37
Total energy for electrical auxiliaries	4.25
B—Steam-Driven Auxiliaries	
1 Boiler room:	
Exhaust heat reclaimed:	
Boiler-feed pumps	1.24
Induced-draft fans (one boiler)	0.02
House service pump	0.01
Exhaust lost:	
Steam jets on burners	0.01
Boiler soot blowers	0.2
Economizer soot blowers	0.43
Superheater soot blowers	0.31
Ash conveyors	0.4
2 Turbine room:	
Steam-jet air pumps	0.36

[1] Estimated from tests.

TABLE 4 COAL CONSUMPTION OF COAL DRIERS

Total coal dried, tons	22,818
Coal used for drying, tons	57
Moisture of coal, drier inlet, per cent	4.54
Moisture of coal, drier outlet, per cent	2.82
Moisture removed, per cent	1.72
Pounds of coal used per ton dried per 1 per cent of moisture removed	2.90

indicating lowest superheater leakage. An accurate estimate of this loss based upon the quantity of make-up places it at 0.9 per cent, for the theoretical make-up required is 2.2 per cent and the actual average was 3.1 per cent. The theoretical make-up of 2.2 per cent has now been reached since steam leakage has been stopped by the use of different material composing the elements.

Loading of Boilers. Figs. 1 and 2 show the average percentage of time during which various ratings are carried, and the expected efficiency with which each of the ratings are carried. Fig. 1 was plotted from actual operating data of November, 1924; Fig. 2 has characteristics derived from observing losses at various ratings and shows a very "flat" efficiency curve.

Tests have indicated 2½ per cent of normal coal feed (at 215 per cent rating) necessary to bank a boiler and maintain its pressure, and operating data reduce the amount below 2 per cent because the proportion of banked and inactive hours to active hours is less than unity. To retain heat in a boiler, all air passages to the furnace are closed immediately upon stopping coal feed, and the unit is practically "bottled" tight so that convection losses are a minimum.

Turbine Loading. Under "uneconomical-loading losses" have been set the weighted losses due to increase of water rate above or below that at the most economical load. It is the practice at the Lakeside station to keep all units, whenever possible, accurately at their most economical loads, these loads being approximately 95 per cent of the full rated generator capacity in the case of the 30,000-kw. units. The load curve illustrated in Fig. 3 shows the greatest throttling loss to occur during the early morning period, 12 to 6 a.m., when approximately 35,000 kw. load was carried by 50,000 kw. capacity. Lakeside's portion of the system load during the period under consideration was limited by a shortage of transmission-line capacity.

The question of load factor influencing economy enters in this connection. That the ratio of average load to maximum-demand load is of smaller consequence in the economy of Lakeside than is the matter of turbine loading is proved by a short calculation. If the midnight-to-morning load is decreased 10,000 kw. and the day load is increased the same amount (see dotted curve in Fig. 3),

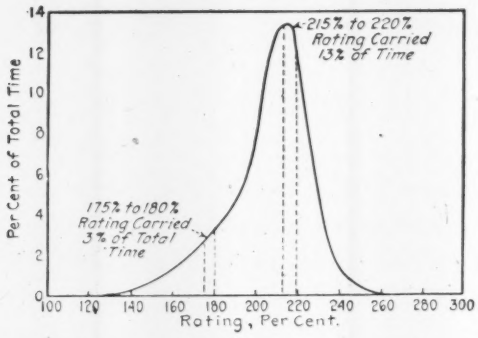

FIG. 1 BOILER LOADING

(Showing percentage of total time that various ratings are carried. Calculated in steps of 5 per cent rating. See notes on curve.)

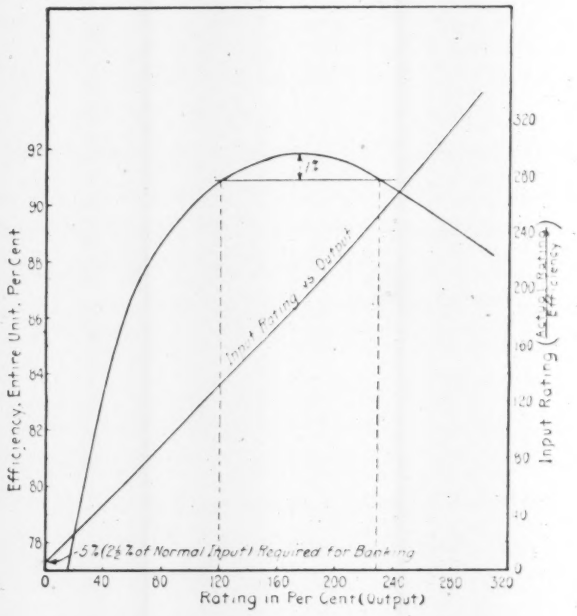
Efficiency, Entire Unit, Per Cent
Input Rating (Boiler Efficiency)
Input & Rating vs Output
2½% of Normal Input Required for Banking
Rating in Per Cent (Output)

FIG. 2 ESTIMATED BOILER EFFICIENCY

(Based upon measured losses observed during actual operation with test instruments. Banking requirements measured by computing from feeder-screw speeds the coal feed necessary to maintain full boiler pressure.)

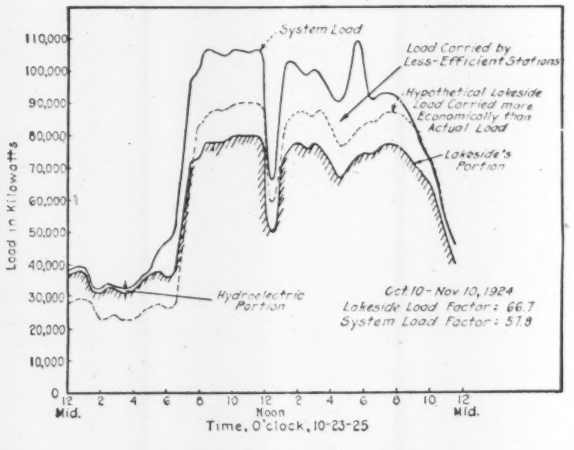

FIG. 3 LAKESIDE LOAD CURVE

(System and Lakeside loads on an average day (October 23, 1924). Lakeside load factor from October 10 to November 10 was 66.7: system load factor during same period was 57.8. The dotted lines show a hypothetical load which the station could have carried ½ per cent more efficiently.)

then banking losses would be increased a maximum of ¼ per cent. Lowering the early morning load for six hours would allow operation of a single 30,000-kw. unit (see Fig. 3), effecting a gain of 0.4 per cent of the day's output. The average full load of 75,000 to 80,000 kw. shown in Fig. 3 was unfortunately above the most economical capacity of the combination of units used (two 30,000-

kw. units and one 20,000-kw. unit), and if increased to 90,000 kw., warranting the starting of another unit, it would have netted an additional gain of 0.3 per cent in the turbine room.

The comparatively greater sustained economy with which pulverized-fuel boilers can carry widely variable loads than steam turbines can carry given loads explains why load factor has not as important an influence upon economy as has the question of turbine loading. During the period considered the load factor of Lakeside equipment was high, but the turbine loading was uneconomical, causing compensatory effects. Economy results obtained after Lakeside carried the entire system load corroborated the above deductions.

Three-quarters of an hour steam consumption at a rate measured by tests is allowed for starting the turbo-generator units.

Unusual Operations. The loss of 0.22 per cent of plant output under this heading is due to testing of emergency governors, setting safety valves, etc.

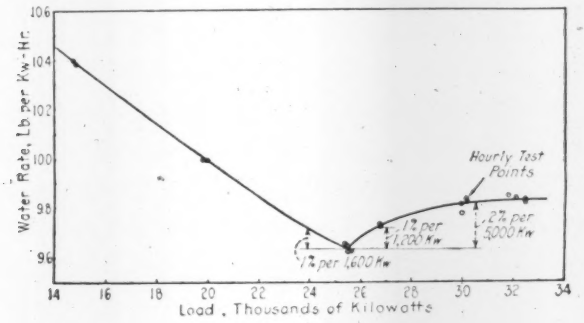

FIG. 4 TURBINE-WATER-RATE VARIATION

(Tests of No. 3 unit, Lakeside, showing the rapid change of water rate at the most economical load, explain why turbine loading may have greater effect upon plant economy than load factor, in view of the low banking losses of the pulverized-fuel boilers.)

Coal Analyses. The average proximate analysis of coal during the period considered is as follows:

	Per cent
Volatile matter	31.4
Fixed carbon	59.4
Moisture	4.02
Ash	9.2
Sulphur	1.1
B.t.u. per lb. as received	13,141
B.t.u. per lb. dry	13,691

The lack of frequent ultimate analyses lends some doubt to the boiler-efficiency calculations in which the losses due to hydrogen burning to water are computed. The analyses possessed average as follows:

	Per cent
Sulphur	1.1
Hydrogen	4.95
Carbon	78.0
Oxygen	6.6

The average fineness to which coal was pulverized during the October 10 to November 10 period was as follows:

Mesh	200	100	50	40	30	20
Per cent through	64	81	96	99.2	99.8	100

The average moisture content at the boiler bins was 1.77, showing that in being pulverized about 1 per cent of moisture was removed.

BONUS SYSTEM

To encourage economy in the use of coal, steam, and electricity, all operators at the station are paid a percentage of the saving they effect in excess of a predetermined standard.

Calculation of the standard station efficiency is made each month, and all operators are paid 45 per cent of the actual coal saved above the standard set. The bonus paid has varied from 8 to 15 per cent of the stipulated wage since the bonus system has been inaugurated.

MINIMIZING OF PLANT LOSSES

At present the air leakage into condensers averages about 5 cu. ft. per min. for each unit. The exhaust pressures maintained in average operation are such that the terminal temperature difference of the exhaust-steam and circulating-water outlet temperatures varies from 11 deg. with 40-deg. inlet water to 8 deg. with 70-deg. inlet water.

Turbine loading, unlike boiler loading, is given careful attention. Except during peak- and low-load periods, the 30,000-kw. units are kept accurately at their most economical loads by making the overload valve gear inactive. A special handwheel is used for this purpose. In such cases the pressure drop through the primary valve is reduced to 7 lb., and is maintained at that point at all times irrespective of slight changes in frequency. Before studying turbine loading, in many instances units were simultaneously operating above and below their individual most economical loads, with losses indicated by Fig. 4.

BOILER-ROOM LOSSES

Maintenance of maximum boiler and economizer air leakage at or below 1.5 per cent CO_2 decrease in the gases has been set as a standard after weekly observation of this source of loss. This leakage includes infiltration through the economizer bottom, a difficult section to keep tight because of water treatment when washing economizer tubes, though this latter infiltration causes no appreciable thermal losses.

Because of the flatness of the pulverized-fuel boiler-efficiency curve and of the small banking losses, various loading combinations show little difference in the resulting economy. Maintenance and other considerations determine ratings of individual boilers rather than economy.

Table 5 gives operating data with boilers operating at 200 per cent of rating or over. Overall station performance was as given in Table 6.

TABLE 5 OPERATING DATA, LAKESIDE PLANT

Plant	Oct. 10 to Nov. 10	Dec.	Jan.	Feb.	March
1 Daily coal consumption, tons*	867	915	934	825	966
2 Daily output, million kw-hr.	1.423	1.528	1.557	1.542	1.629
3 Make-up water, per cent	3.12	3.80	3.19	2.85	2.51
Boiler Room					
4 Rating	213	227	228	228	225
5 Boiler hours, daily	141.7	136	138.5	136.94	142.4
6 Banked hours, daily	17	18.1	19.5	19.3	28.63
7 Combustion-air temperature, deg. fahr.	99	82	84	84	83
8 CO_2, per cent	14.7	14.8	14.8	14.8	14.7
9 Steam temperature leaving superheater, deg. fahr.	702	687	710	721	717
Gas Temperatures, Deg. Fahr.					
10 Economizer inlet	564	572	573	574	568
11 Economizer outlet	232	237	243	244	248
12 Drop in temperature	332	335	330	330	320
Feedwater Temperatures, Deg. Fahr.					
13 Economizer outlet	251	248	254	258	260
14 Economizer inlet	119	121	133	137	137
15 Rise in temperature	132	127	121	121	123
Turbine Room					
Temperatures, Deg. Fahr.					
16 Throttle, Nos. 1 and 2 units	671	647	640	....	619
17 Throttle, No. 3 unit	682	660	677	683	684
18 Throttle, No. 4 unit	689	658	688	701	695
19 Throttle, No. 5 unit	....	673	693	702	701
20 Condenser hotwell	73	63	64	63.2	61.8
21 Circulating-water outlet	67	53	56	55.9	54.4
22 Circulating-water inlet	53	38	41	41.1	40.7
23 Circulating-water rise	14	15	15	14.8	13.7
Pressures, Lb. Gage and In. Hg Abs.					
24 Throttle, all turbines	288	286	289	291	293
25 Exhaust Nos. 1 and 2 units	0.91	0.63	0.56	....	0.56
26 Exhaust Nos. 3 and 4 units	0.94	0.78	0.76	0.76	0.74
Preparation House					
27 Per cent moisture, drier inlet	4.54	5.58	6.75	7.64	7.18
28 Per cent moisture, drier outlet	2.82	2.52	2.48	2.85	2.72
29 Per cent moisture removed	1.72	3.06	4.27	4.79	4.46
30 Tons coal used in drier daily	1.84	6.08	6.96	7.56	8.23
31 Lb. coal used per per cent moisture removed per ton dried	2.90	4.72	3.53	3.41	3.82

* Approximated from daily electrical output. Accurate coal tonnage determined only bi-weekly.

TABLE BOILER AND STATION PERFORMANCE DURING THE SIX MONTHS ENDING APRIL 1, 1925

Month	Plant heat consumption, B.t.u. per kw-hr.	Boiler efficiency, steaming only, per cent
October, 1924	16,000	89.71
November, 1924	15,945	91.34
December, 1924	16,308	89.67
January, 1925	16,066	88.85
February, 1925	15,861	90.05
March, 1925	15,930	[1] 89.7

[1] Reduced due to greater use of oldest equipment.

Conclusions

A modern power station may be so designed and operated that its ultimate efficiency will be within 5 per cent of the combined efficiency of its component equipment working at most economical load.

The combination of boilers, turbines, and auxiliaries, allowing for radiation and miscellaneous losses, may work at an efficiency ratio of 64 per cent in a pulverized-fuel station. This percentage multiplied by the thermal efficiency of the cycle employed and by 95 per cent operating efficiency ratio should give the approximate ultimate overall thermal efficiency of any pulverized-fuel station using the usual equipment.

Totalizing the losses of energy in the Lakeside station, all of which except those of radiation and a few minor ones are measured, and applying the total to the theoretical efficiency of the cycle employed, shows that a performance of 15,800 B.t.u. per kw-hr. is reasonable.

Decrease in boiler-unit efficiency below that at most economical rating due to operation throughout the range of loads found in actual practice may aggregate less than one-half of one per cent in the case of a pulverized-fuel installation.

Banking losses of a pulverized-fuel station operating at 65 per cent load factor may be limited to 2 per cent.

The energy cost of pulverizing coal was 1.1 per cent, and that of conveying and pumping it before and after pulverization was almost negligible.

Drying the coal before reaching the furnaces consumed 0.25 per cent of the total coal flow. Assuming 30 per cent drier efficiency and allowing for the heat necessary to evaporate this moisture in the furnace if not previously dried, the net energy cost of the coal-drying process was 0.16 per cent.

Of the losses in the process of combustion in the furnace and of heat absorption in the boiler, superheater, and economizer, 45 per cent are unavoidable because of hydrogen and moisture in the coal and air. Allowing for these unavoidable losses, the boiler unit may be considered to have an efficiency of 95.4 per cent.

Load factor in the case of a pulverized-fuel station may have less influence upon coal economy than has the question of turbine loading. During the period considered in the paper, Lakeside had a high load factor, but was handicapped by having to carry loads of such certain magnitude that the economy of turbine loading was poorer than under average conditions.

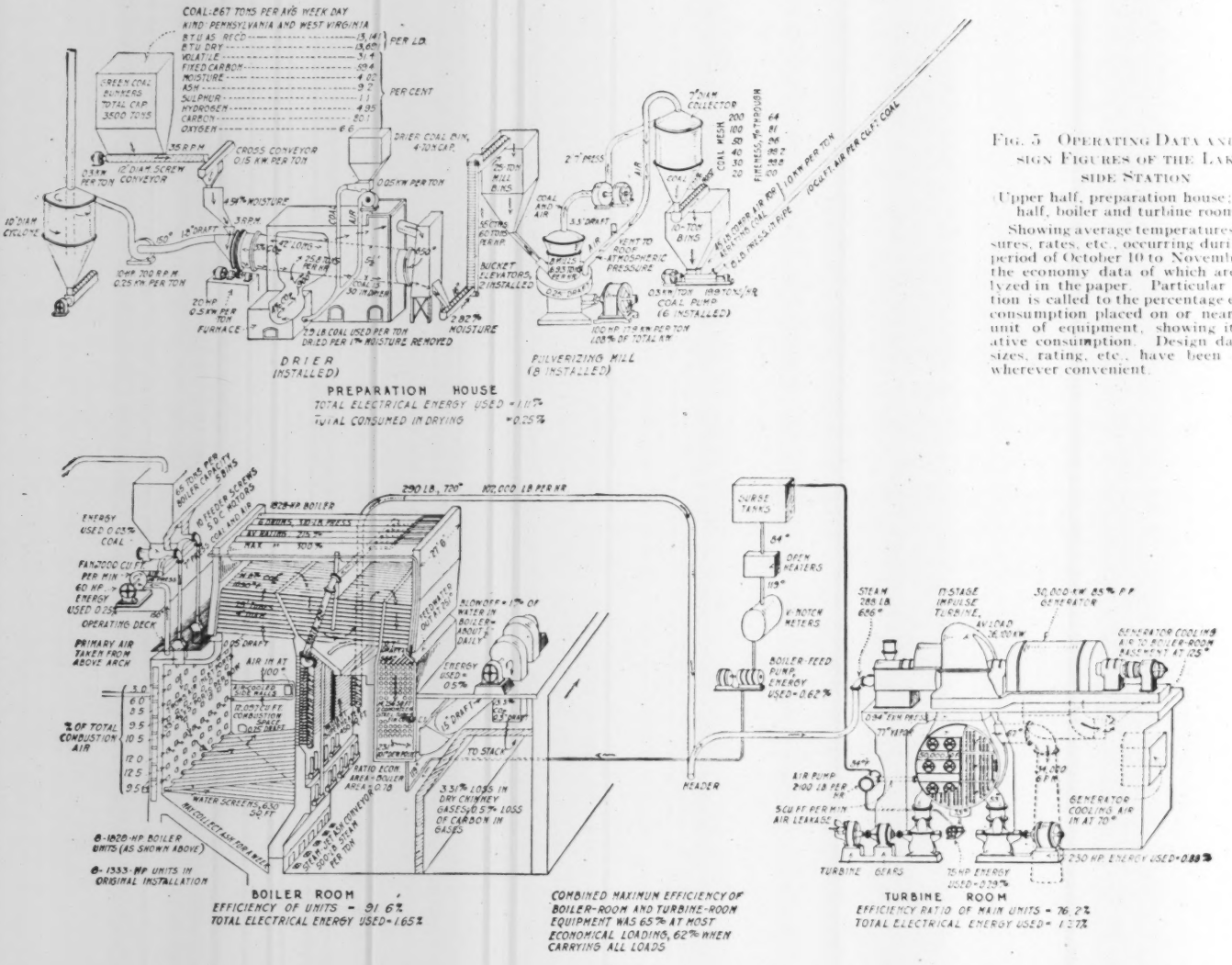
PREPARATION HOUSE
DRIER (INSTALLED)
PULVERIZING MILL (8 INSTALLED)
BOILER ROOM
EFFICIENCY OF UNITS = 91.6%
TOTAL ELECTRICAL ENERGY USED = 1.65%
COMBINED MAXIMUM EFFICIENCY OF BOILER-ROOM AND TURBINE-ROOM EQUIPMENT WAS 65% AT MOST ECONOMICAL LOADING, 62% WHEN CARRYING ALL LOADS
TURBINE ROOM
EFFICIENCY RATIO OF MAIN UNITS = 76.2%
TOTAL ELECTRICAL ENERGY USED = 1.27%

Fig. 5 Operating Data and Design Figures of the Lakeside Station

(Upper half, preparation house; lower half, boiler and turbine rooms.)

Showing average temperatures, pressures, rates, etc., occurring during the period of October 10 to November 10, the economy data of which are analyzed in the paper. Particular attention is called to the percentage energy consumption placed on or near each unit of equipment, showing its relative consumption. Design data of sizes, rating, etc., have been added wherever convenient.

Watt's Garret Workshop

THROUGH the generosity of Major Gibson Watt, the English nation has recently come into possession of the famous garret workshop from Heathfield Hall, in which Watt spent many hours of his declining years. Heathfield Hall was built by Watt in 1790, and stands not far from Handsworth Church. The Hall and its grounds for long remained comparatively unaltered, and last year during the World Power Conference an attempt was made to secure the house as a memorial to Watt. The scheme, however, fell through, and the land will soon be in the hands of the builder. But the workshop and the whole of its contents, with most of its fittings, have been given to the Science Museum, where a replica of the room has been constructed. Before being dismantled the workshop and its shelves, benches, lathe, tool chests, busts, etc., were completely catalogued and photographed, and these will all be arranged in the room in the Science Museum as they were presumably left by Watt.—*Engineering*, July 31, 1915, p. 141.

Steam Logging

Development of the Steam Logging Engine and Its Use on the Pacific Coast—Methods of Logging in General Use, and Types of Engines Applied to Them

By JOSEPH W. GILL,[1] PORTLAND, ORE.

THE USE of steam in connection with logging on the Pacific Coast became a recognized factor as early as the year 1890. Previous to that time only what might be termed "makeshift" machines were used, which, however, according to some of our early loggers, were able to put logs at the landing as economically as some of the larger and heavier engines of today, that is, for the length of haul and other conditions under which they had to work.

Among the early types were the vertical gypsy, and the Dol Beer. The first of these consisted of a vertical boiler with a single-cylinder vertical engine secured to the boiler and running through a bevel gear and pinion to a vertical shaft on which a gypsy was mounted; this was first used about 1885 in connection with ground-yarding.

The Dol Beer, getting its name from John Dolbeer of Salmon Creek, near Eureka, California, who first conceived the idea of putting it on a sled, was of slightly different design. This machine, as shown in Fig. 1, consisted of a vertical boiler and a single-cylinder engine with a heavy pulley on the crankshaft for a flywheel and a gypsy on either end of a main shaft driven by a gear and pinion from the crankshaft. The two vertical rolls in front were used for a fairleader to guide the line on to the gypsy when the line came from either side of the machine.

These machines were operated at a boiler pressure of not over 100 lb. Another type used was a single-drum engine of the pile-driver type, often an engine which had been discarded for its original purpose on account of some deficiency. With the single drum, as with the gypsy, it was necessary to use horses to haul the line back to the point where the tree was located from which the logs were being cut.

As the hauls became longer it became necessary to speed up the operation, so the horses were discarded and a second drum was added which operated a line to haul the main line back to the cutting. This line is commonly called the "trip line."

The present development of the logging engine began soon after 1900. At that time the most popular sizes were the 9 by 10-in. and 9½ by 11-in. double engines with wide drums which were used as road engines on long yarding operations.

These engines weighed about 19,000 lb. and had cable capacities on the main drum of about 2000 ft. of 1-in. line and double this amount of ½-in. line on the trip drum. The boiler was a 48-in. or 54-in.-diameter vertical type, working at 125 lb. pressure.

The line speeds were approximately 400 ft. per min. on the main line and 450 ft. on the trip. At this time these engines were in use for yarding and roading, both on short and long hauls. As the roads began to reach out further it became necessary to go to a slightly larger cylinder size and more drum capacity.

The 10 by 13-in. and the 11 by 13-in. were the next step, these engines weighing from 31,000 to 36,000 lb. and having line speeds of about 300 ft. per min., giving a higher pull relatively, and cable capacities of 4100 ft. of 1-in. and 6900 ft. of ⅝-in. line.

A "yarder" developed at about this time was a type known as the "Crackerjack," having two drums mounted on the same shaft. This engine was first made in the 9½ by 10-in. size and later increased to 10 by 13-in., had a 48-in. boiler working at 125 lb. Its capacity was 1500 ft. of 1-in. line, weight about 20,000 lb., and the line speed about 450 ft. per min., or slightly faster than the road engine of that date.

From 1904 to 1906 the sizes increased rapidly to 11 by 13 in., 12 by 13 in., and 13 by 13 in. Sizes as large as 14 by 14 in. with a 72-in. extended-firebox boiler, working at 200 lb. and with a line capacity of 8100 ft. of 1¼-in. line and 21,400 ft. of ⅝-in. trip line, have been used with speeds of 650 to 900 ft. per min.

The long-haul road engine is gradually dropping out, it having been found more economical to build rail lines in the timber and yard to them. In some instances it became necessary to relay the logs; in this arrangement a second yarder was used, the one nearest the rail line being called a "swing" engine.

About 1905 a special engine, the first of its kind, was developed by W. H. Corbett, at that time the president of the Willamette Iron & Steel Works, known as the "Humboldt" yarder, to haudle the heavy, large logs in the redwood country. This engine dne to compound gearing was quite slow in line speed, running from 285 to 425 ft. per min. on the main drum.

In operations where higher speeds could be used this engine has not been so satisfactory and engines of two-speed type have been developed. These are of several designs, the first, Fig. 2, being perhaps the type having tandem wide drums, with the train of gears on the main shaft, one on either side of the main drum, commonly called the simple-geared two-speed, one set of gears giving a line speed of about 518 ft. and the other about 1090 ft. per min. These were first used in what is called the "high-lead" operation.

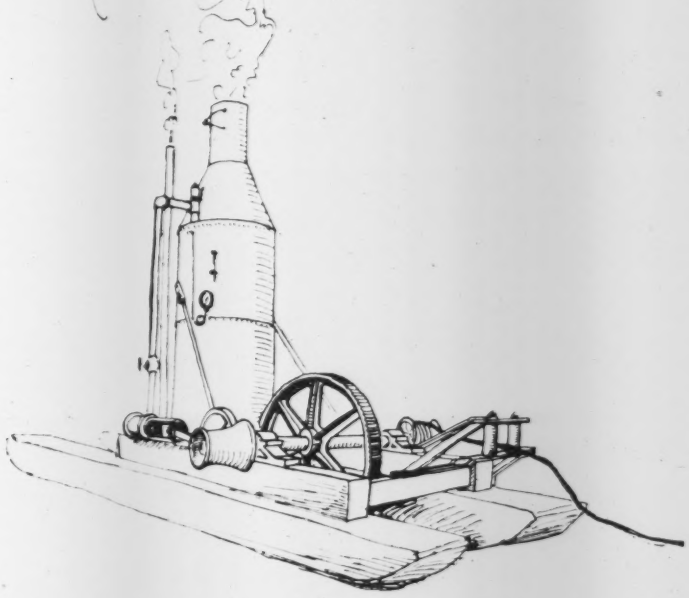

FIG. 1 THE DOL BEER LOGGING ENGINE

Other engines of the two-speed type are equipped with compound gearing, with either instantaneous gear change or the change being accomplished by the use of sliding jaw clutches of different designs. The latter, in most instances, has worked out to very good advantage, the low gear being used to move the engine into its setting; when in position and the change is made into high gear, it is left for some considerable time, that is, until it is desired to again move the engine.

The latest type of these engines is built in 12 by 13-in., 13 by 13-in., and 14 by 14-in. sizes with steam pressures up to 225 lb. Some of these machines have as many as five drums, depending upon the kind of operation in which they are to be used. They weigh from 65,000 to 117,000 lb. and are frequently mounted on steel cars as they are often used alongside the rail lines. (See Fig. 3.)

Changes other than of type and size were taking place at the same time, first among them being boiler types and working pressures. The working pressures were increased from 125 lb. to 175 lb., and about 1910, pressures of 200 lb. were used. This of course called for heavier or stronger parts throughout. Steel

[1] Chief Designer, Smith & Watson Iron Works, Mem. A.S.M.E.

Presented at the Portland Regional Meeting of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, Portland, Ore., June 22–25, 1925. Abridged.

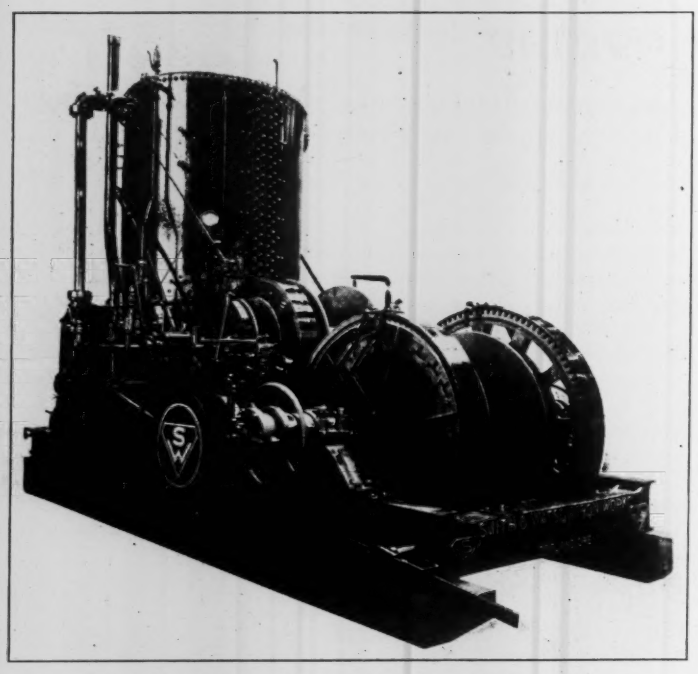

FIG. 2 SIMPLE GEARED TWO-SPEED YARDER

frames and bearing caps, nickel steel for crankshafts, etc., were embodied in the engine designs to meet the increasing loads without undue weight increase.

The Smith & Watson Iron Works have adopted 225 lb. working pressure as their standard, and while this pressure is not used by all the operators it gives a much longer-lived boiler and the capacity is there in ese should it be wanted for any extrahard duty.

The round-firebox boiler of the early types was superseded by the single extended-firebox boiler, and this in turn by the double extended-firebox type. This latter construction gives a very large furnace and combustion chamber and a larger grate area in proportion to heating surface.

Years ago it was common to use the best logs for fuel, as they were easier for the wood bucker to split, but in many camps at present as economy is being practiced the fuel is often not of the best. A trial is being made of cleaning out the old cuttings and hollows, gathering the old tops and discarded logs which were of no use as lumber and using them for fuel. This type of fuel is not the best there is, and it is desirable therefore to have a large body of fire.

In the double extended-firebox boiler as built by the Smith & Watson Iron Works there is an exceptionally high furnace, this being accomplished by dropping the grate lower by a specially designed dead plate. (Fig. 4.) This dead plate also prevents the cold air from rushing up the sides of the furnace.

Piston valves have been adopted by nearly all of the builders, feedwater heaters are used, and even superheaters have been suggested. Other details such as the friction devices have passed from the hand-operated screw type to steam-operated mechanisms both of the simple direct-acting, and the toggle double-acting types. On skidders air-operated devices have come into pretty general use, and on the larger machines brakes are operated by compressed air.

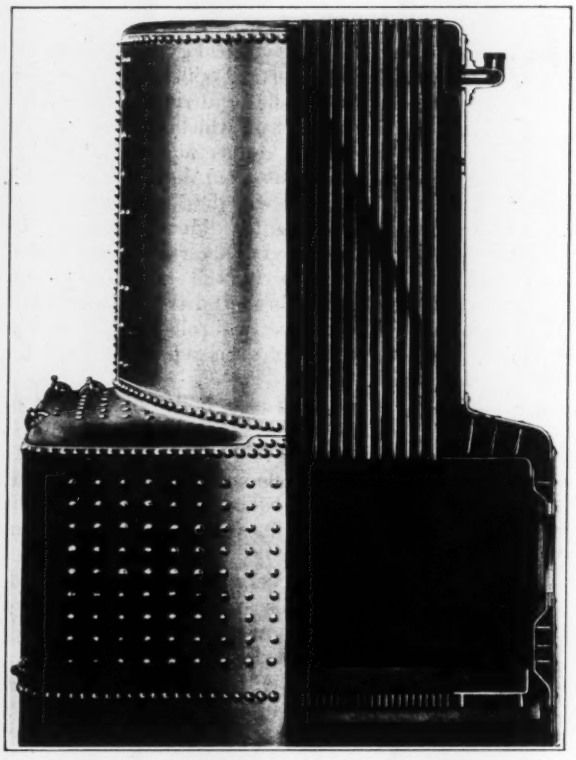

FIG. 4 HALF-ELEVATION AND HALF-SECTION OF DOUBLE EXTENDED BOILER SHOWING LOW GRATE

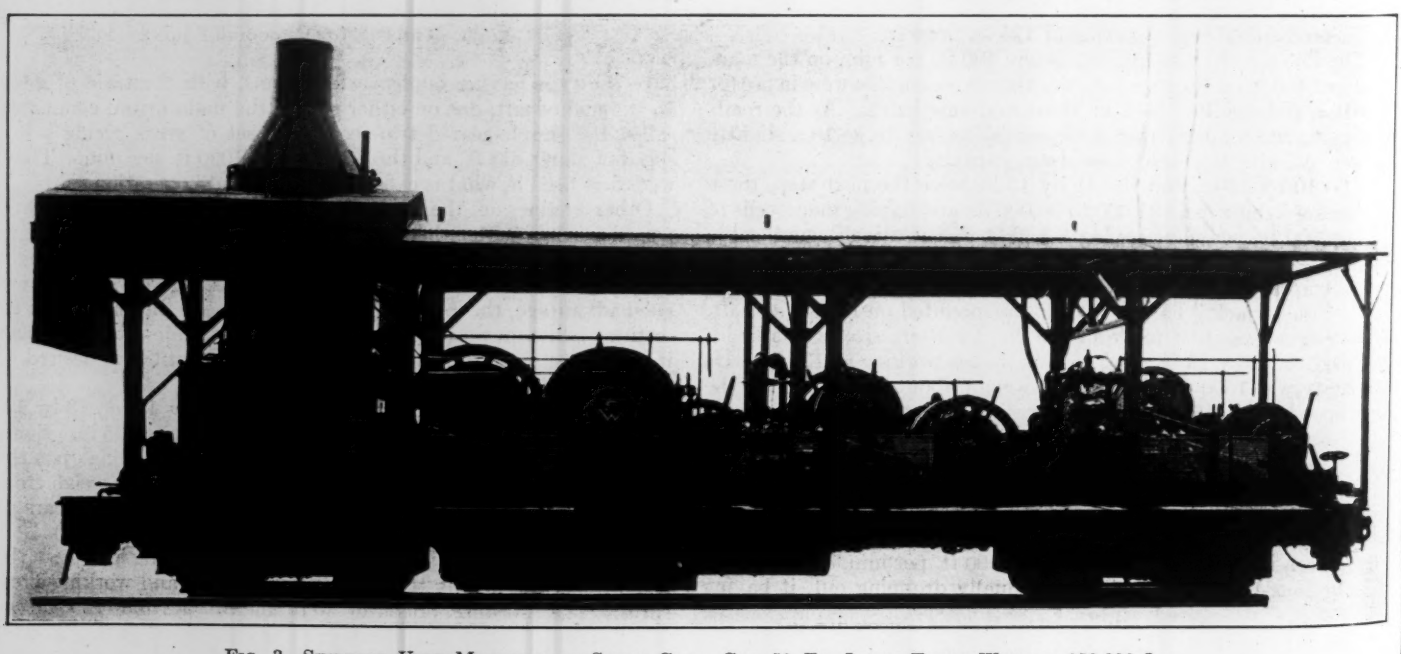

FIG. 3 SKIDDING UNIT MOUNTED ON STEEL CAR. CAR 52 FT. LONG. TOTAL WEIGHT, 250,000 LB.

LOGGING SYSTEMS

The principal systems employed by the western logger are ground yarding and roading, high-lead, and aerial systems.

The high-lead system, which may be called "semi-ground yarding," is used to best advantage on short yarding operations, otherwise there is no real benefit to be gained by the high lead.

The aerial systems may be divided into two distinct classes; the slack skyline, in which the overhead line is raised and lowered at each trip of the trolley, and the tight or standing line, in which the overhead line remains taut and the hook or fall block is pulled to hooking position by the trip line to either side of the road. This perhaps has some advantages over the slack skyline as the hook tenders are not endangered by the falling of the skyline.

ROADING AND GROUND YARDING

In the past the common practice was to haul the timber long distances over pole roads. These roads ranged from 2000 to 6000 ft. up to even 12,000 ft. in length. Six to eight logs were hauled in one train, being fastened together one behind the other. In some camps the choker was attached to the last log of the train and this was used as a pusher.

The engine used in roading and ground yarding is of the tandem drum type. Sizes range from 10 by 12 in. with cable capacities of

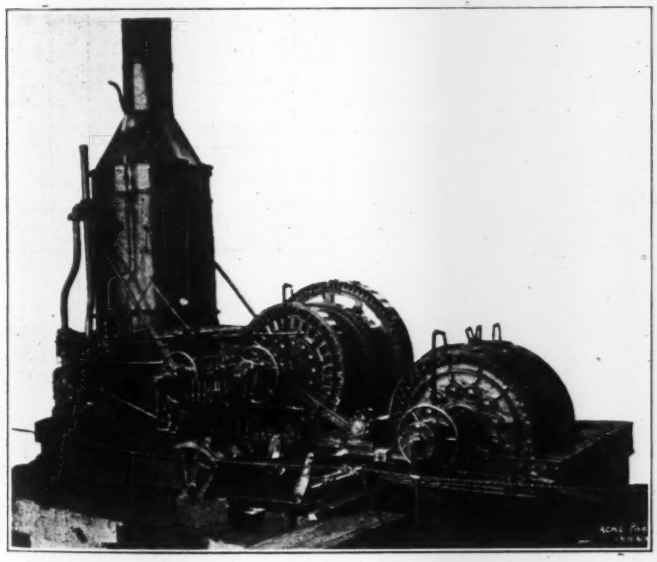

FIG. 6 13 BY 13-IN. TWO-SPEED SKYLINE ENGINE

4000 ft. of $1^1/_8$-in. line on the main drum and 8000 ft. of $^5/_8$-in. line on the trip drum and weighing about 35,000 lb., to 14 by 14-in. engines with cable capacities of 8000 ft. of $1^1/_4$-in. line on the main drum and 16,000 ft. of $^3/_4$-in. on the trip drum and weighing 70,000 lb. Line speeds run from 550 ft. on the main line to 900 ft. on the trip line. Boilers range from 54-in. round up to 78 in. round. This general type of machine is still used by small operators in certain advantageous "shows," and produces logs economically on level or easy sloping ground.

HIGH-LEAD YARDING

For high-lead yarding the same equipment, as far as engines are concerned, is used as for ground yarding. The main lead block is attached near the top of the spar tree as high as possible, from 150 to 200 ft. from the ground, this giving a lifting effect to the hauling line. This is the chief advantage of this system and allows the logs to travel much faster as they are not stopped by stumps and other obstructions. (See Fig. 5.)

AERIAL SYSTEMS

In order to eliminate the excessive loss through breakage of timber and heavy line costs resulting from ordinary ground or high-lead yarding systems it has become necessary to adopt aerial systems of various kinds. This is particularly essential where the ground is rough and railroad construction expensive. Several systems have been developed for this purpose, notably the Lidgerwood, McFarlane, and North Bend. The Lidgerwood system is fully described in a paper by Spencer A. Miller.[1]

McFarlane System. In this system the engine is a four-drum machine having a drum for tightening and reeling the standing line, a main or haul-in drum, a trip drum to handle the line used to haul the carriage back to the tail tree, and a straw drum on which is reeled the line used to handle the other lines when rigging and changing to new tail trees.

Several engines of slightly different arrangement of drums have been designed to handle this system, one of which is shown in

FIG. 5 ARRANGEMENT OF LINES FOR HIGH-LEAD YARDING

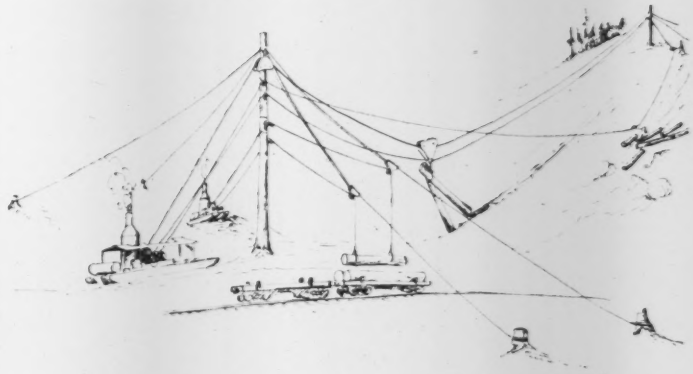

FIG. 7 ARRANGEMENT OF LINES FOR MCFARLANE SYSTEM

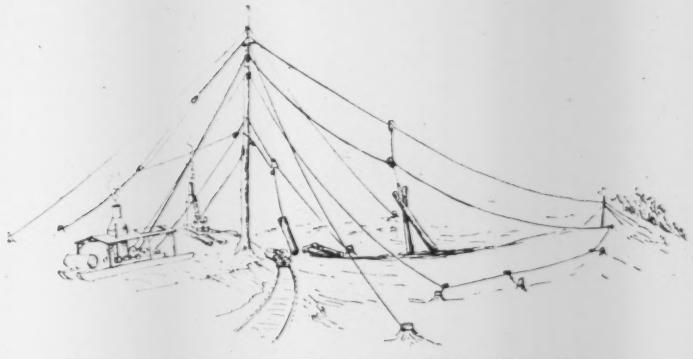

FIG. 8 ARRANGEMENT OF LINES FOR NORTH BEND SYSTEM

Fig. 6. This machine is in use in California and is handling large redwood logs down very steep hill sides. The skylining drum holds 1800 ft. of $1^1/_2$-in. line, the main-line drum 2350 ft. of $1^1/_4$-in. line, the trip drum 3360 ft. of $^7/_8$-in. line, and the straw-line drum 4200 ft. of $^7/_{16}$-in. line.

This system was first used by C. E. McFarlane, now of Seattle, Wash., in 1905, in yarding logs down a steep slope. The rigging consists of a main cable suspended between a head tree and a tail tree. On this line the carriage travels. The main cable leads from the drum on the engine and passes through a

[1] The Overhead-Cableway Method of Logging, MECHANICAL ENGINEERING, July, 1925, p. 527.

large block near the top of the head tree and from there to a shoe on the tail tree; passing through the shoe it is made fast to a stump or tree. (Fig. 7.)

All engines of this type must have large drum capacity and tremendous power for moving and for raising the skyline and log in direct pull from the drum.

This system has been used successfully to yard logs at a distance of 2500 ft., the distance between trees depending on the character of the ground and distance to be yarded.

The skyline is raised and lowered at each end of the haul in order to fasten and unfasten the choker or loop of cable which passes around the log to the carriage. The raising and lowering of the skyline for each load is the outstanding feature of this system.

North Bend System. Credit for this system is given to R. W. Vinnage of North Bend, Wash. It is used in many camps in both Oregon and Washington and its popularity is due to its simplicity and the fact that it may be operated with an ordinary ground or high-lead yarder with tightening drum attached.

In this system the skyline remains in a fixed position and is called the "standing line" for this reason. It is held by a system of blocks, the skyline itself being fastened to one of these blocks. The line for tightening is smaller in size, usually 1 in. in diameter, and passes from a small drum in front of the engine to the head tree and from there to the block. (Fig. 8.)

The main line passes from the drum through a block on the head tree, through the fall block and is attached to the trolley. The trip line passes from the drum through a block on the head tree and a series of blocks along the side of the road to the tail tree and back to the fall block.

As Fig. 8 shows, the movement of the trolley and the raising or lowering of the log to suit the contour of the ground are controlled by the trip line.

As a practical logging question it is not always clear just which of these systems is the best and one operator will get good results with ground yarding and another with practically the same conditions to work against will do the work with high lead. Quite often the method is influenced by the machines which the operator has at hand and the amount of timber to be logged may not justify the purchase of special equipment.

Lowering Engines

The lowering or incline engine has also taken a prominent part in helping to solve some of the difficult problems of the Pacific Coast logger. In certain places as many as 15 miles of expensive railroad construction which would have been necessary have been eliminated by the installation of one of these engines. They have been successfully operated on inclines from 4000 to 7000 ft. with maximum grades of from 50 to 65 per cent.

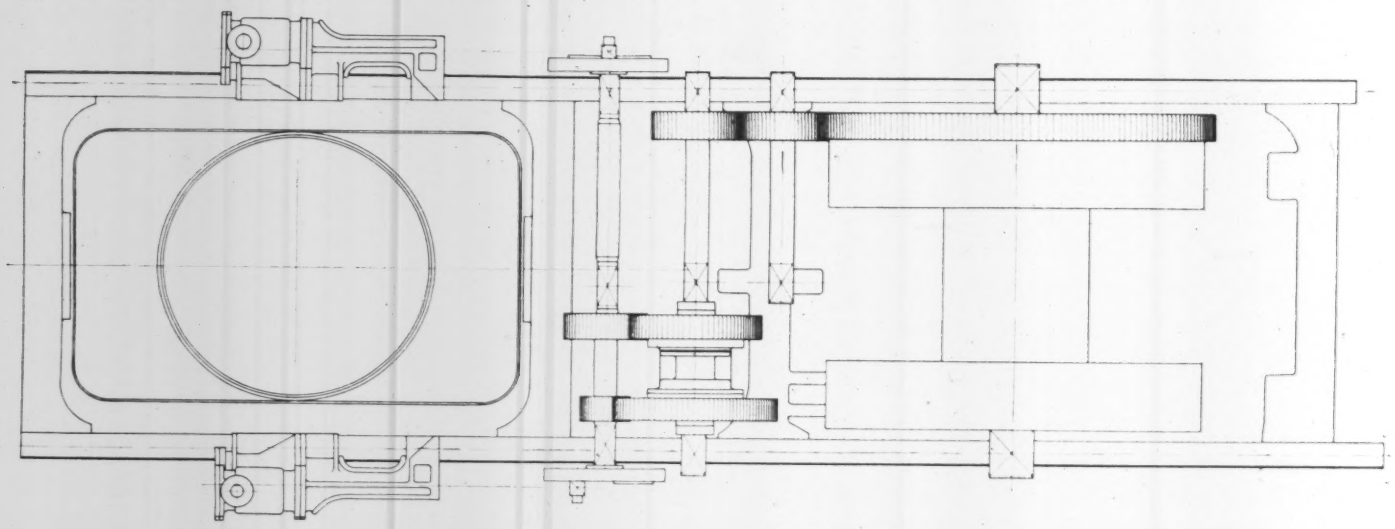

FIG. 9 DRUM ARRANGEMENT, 13 BY 13-IN. TWO-SPEED LOWERING ENGINE

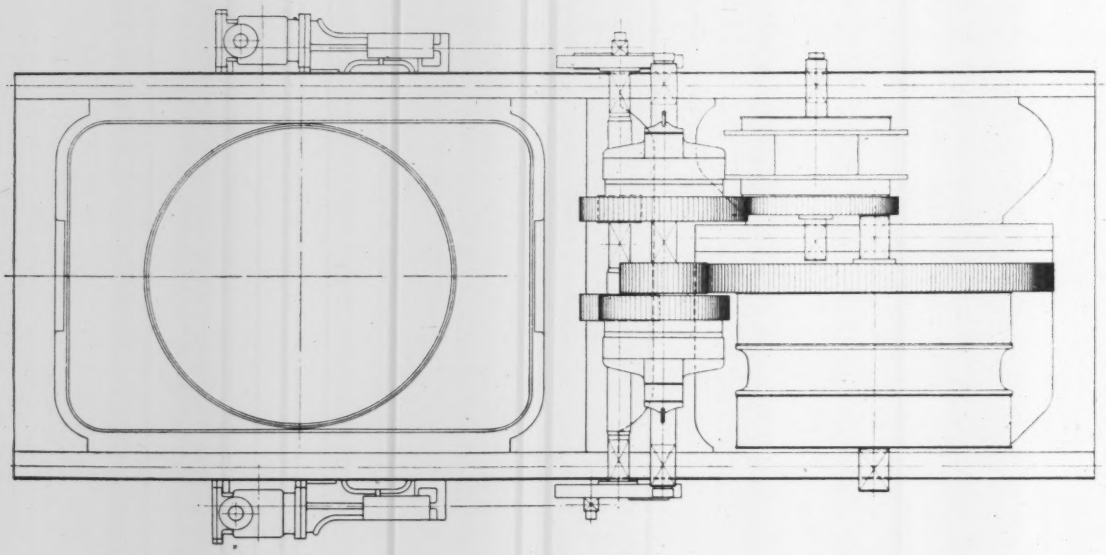

FIG. 10 DRUM ARRANGEMENT, 13 BY 13-IN. TWO-SPEED COUNTERBALANCE LOWERING ENGINE

The three principal systems used are the direct, counterbalance or gypsy system, and the Sessoms system.

The Direct Type. The first engine for the direct system which was used in Oregon or Washington was built by the Willamette Iron & Steel Works about 1904. The cylinder size was 16 by 20 in. and steam was furnished by a stationary boiler of the locomotive type. Considerable difficulty was experienced in getting the proper amount of braking surface and this was greatly increased after the engine was put in operation.

It was used near Rainier, Ore., for a number of years on an incline about 5000 ft. long. Later it was taken to Greenleaf, Wash., where it is still in use on an incline about 6340 ft. long, with a maximum grade of 66 per cent at the bottom to 12 per cent at the top. Twenty-four cars are handled in eight hours with an average log scale of 8000 ft. to the car. Six minutes are required for the down trip.

Many engines of this type have been built in the last five years. Among the largest of these was one manufactured by the Smith & Watson Iron Works and now in use in British Columbia on an incline nearly 7000 ft. long with a maximum grade slightly over 45 per cent. This machine has 13 by 13-in. cylinders and a 72-in. double extended firebox boiler operating at a working pressure of 225 lb.

As shown in Fig. 9, there is a double set of gearing between the crankshaft and the intermediate shaft, giving mean speeds of 300 to 700 ft. per min. on the drum. The low gear was used in hauling the engine up to its setting over very rugged, mountainous country. In places it was so steep that it was necessary to hold the engine with cables while the blocks were being changed.

Another engine of this type has lowered 40 cars of logs scaling 300,000 ft. in one day over an incline 2700 ft. long.

Counterbalance or Gypsy System. The engine for this system, as the name implies, has on what would ordinarily be the drum shaft a drum with the form of a large gypsy. (Fig. 10.) The line makes three and one-half turns on the gypsy and the ends are attached to cars. As one car comes up empty, a loaded one goes down, the weight of the empty car and cable coming up serving as a counterbalance for the loaded one going down.

One of these engines which was built in 1923 by the Smith & Watson Iron Works has been in operation at Brinnon, Wash., on an incline 6400 ft. long, said to be the longest incline of this type in Oregon or Washington. The grade varies from a minimum of 14 per cent to a maximum of 50 per cent.

The gypsy is 80 in. in diameter, made of manganese steel in six segments and using a $1^1/_2$-in. Roebling cushion-center cable. The cable is 7000 ft. long in one piece. The time of lowering a load of logs scaling 10,000 ft. having a weight of 120,000 lb. including car, is six minutes, and the load is under perfect control during the whole trip. This time could be decreased, if necessary, without damage to the engine.

Sessoms System. This system uses the same type of engine as the direct system, the only difference being in the cable capacity of the drum. On an incline where it would be necessary to use $1^3/_4$-in. cable with the direct system it is possible to use a $1^1/_4$-in. cable of twice the length with the Sessoms system.

In the Sessoms system the cable passes from the drum on the engine to a specially designed car carrying one large sheave 6 ft. in diameter, around which it passes and then runs back to a point at the side of the engine where the dead line is made fast. A swivel is placed near the end of the line to take care of the twist.

To the car, the feature of this system, is attached the loaded car or cars, sometimes as many as four, depending on the grade and size of load. A series of line guides consisting of posts, at close intervals, on top of which are placed grooved rolls, is located along either side of the road so that at no time does the cable touch the ground. The outrigger arm of the car passes directly over the guides and as the car moves the cable is either placed in or taken up from the sheaves on the guide posts. Special guides with trips and arms for holding the cable are placed at curves.

Engines of all these types are equipped with reversing valve gear of the outside type, a modified Marshall gear being used by most of the builders. This gives more space on the inside of the frame for gear change mechanism.

The dissipation of heat generated by the brakes has been one of the great problems on all these engines, especially on the longer and steeper inclines. Splash systems have been used unsuccessfully, the main trouble being the wetting of the gears and mixing water with the grease on them, also the wetting of the braking surfaces. Guard plates have been placed over the drum and gear, but this does not seem to remedy the fault.

The Smith & Watson Iron Works designed and built into four of its lowering engines two years ago a new method for taking care

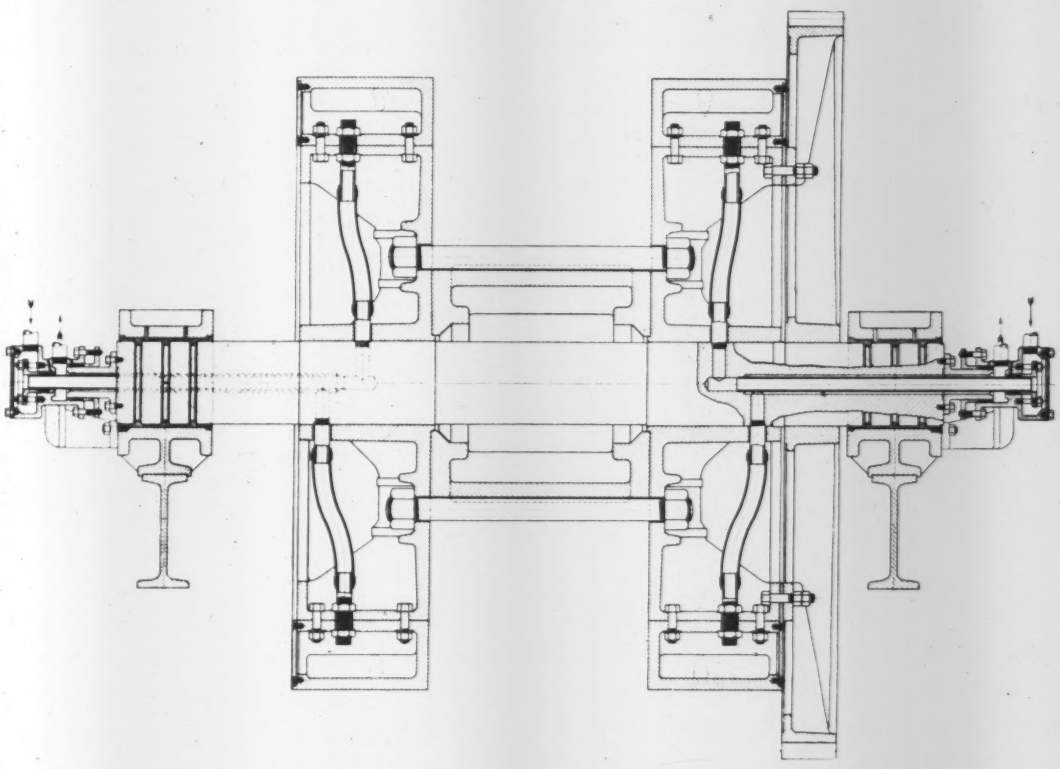

FIG. 11 METHOD OF CIRCULATING COOLING WATER IN LOWERING-ENGINE BRAKES

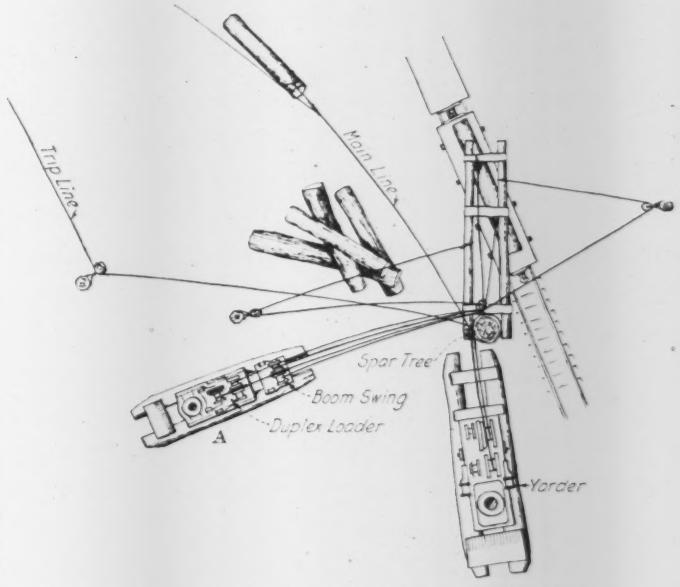

FIG. 12 ARRANGEMENT OF BOOM FOR LOADING

of this water. As shown in Fig. 11, water passes through the connection on the end of the shaft, through the pipe and shaft to the chamber on the rim of the drum and is returned to the shaft and out through the connection on the end. This gives a positive method of circulation and there is no leakage or loss of water. The chambers are made of cast iron, secured to the drum flanges, and have a series of webs cast in, giving a large cooling surface.

Compound hand levers were used on the earlier engines for setting

brakes. On the later types compressed air has been used and has been found to answer the purpose much better than steam. Some builders equip their engines with brakes set automatically by weights and the air or steam is used to release rather than set them. Hand levers are provided to release the brakes in case the engine is not in operation, but, as is readily seen, in case anything happens to the releasing gear, the brakes set themselves.

LOADING

As it is possible in many locations to yard logs faster than they can be loaded, the logger has found the loading of logs one of his most difficult problems. This perhaps accounts for the many different systems in use. However, during the last few years the duplex system has been found most easily adapted to all conditions.

With ground yarding, where the logs may be landed parallel to the track, other systems are used, called the "crotch" line and skyline systems, but where the logs come in in almost any position and land in a heap as they do in high-lead yarding and the skyline and skidder operations, the duplex is most efficient.

This system is also being used in connection with a boom, Fig. 12, which makes a very satisfactory arrangement as the loading tongs may be swung to a position over the pile of logs, thus saving much time which would otherwise be used in dragging the heavy tongs about.

For the duplex system a machine with double set of reversing engines is used; $9^1/_2$ by 10 in. is the popular size, though they are built in 10 by 12-in. and 11 by 13-in. sizes. The drums are fast to the shafts, doing away with friction devices, the position of the log being at all times under control of the reversing valve levers.

As Fig. 10 shows, blocks are hung on guy lines and centered over the track. The lifting lines lead through these blocks to the spar tree and from there to the loader drum.

When the log is raised clear of the ground it automatically comes to a position over the center of the car, when it is only necessary to lower it to the load.

Another system which is considered very fast by some operators is known as the single-tong system. (Fig. 13.) As the name implies, there is only a single set of tongs which are used in lifting the log, the line passing from them directly to the drum, through a block suspended from a guy line and over the center line of the track.

This system requires a very skilful operator, and unless handled very carefully the swinging logs are apt to damage the cars.

The difficulty with this as well as with the duplex when blocks are suspended from the guys lies in the swinging of the tongs far enough from the track to fasten them to the log to be lifted.

Unless the logs are landed close to the track it sometimes takes several minutes to attach the tongs to a log, thus holding up the yarding operation.

The combination of the boom and the duplex engine previously mentioned overcomes this difficulty to a great extent.

Electric Logging

BY P. A. WICKES,[1] PORTLAND, ORE.

IN REVIEWING the progress of logging by electricity, which has made such advances in recent years, we find that the first experimental machines were built about fifteen years ago and proved quite satisfactory, considering that many of them were steam donkeys converted by removing the boiler and prime mover parts and substituting therefor an electric motor with suitable gearing to connect to the old drum system. A few of the early operators of these machines were the Potlatch Lumber Company in the white-pine timber, and Smith Powers Logging Company, and the C. A. Smith Lumber and Manufacturing Company in the fir timber. The horsepower of the a.c. motors used ranged from 150 to 265.

The majority of these machines were used in moderately light timber and had much lower cable speeds than present-day machines, the relative cable speeds of the two designs being approximately as follows, based on average drum diameter: Original machines, main line 300 ft. per min., trip line, 850 ft. per min., as compared to the present-day machine which has main-line speeds of 375 ft. per min. in low gear and 775 ft. per min. in high gear; the trip line speed being 900 ft. per min. in low gear and 1900 ft. per min. in high gear.

From 1913 to 1917 there was very little activity along electric-logging lines, and naturally very little new development. In the latter year, however, the Snoqualmie Falls Lumber Company, a large fir operator, converted a steam yarder to electric drive, using a 200-hp. a.c. motor; this machine was employed for some time in logging heavy timber. In 1920 the company installed an electric loader with one side[2] completely electrified, and from performances and tests of these two machines they decided to electrify their entire wood-logging operation in 1921. This was the first commercial electric-logging operation attempted in the Pacific Northwest.

Building on the data and experience gathered from 1911 to 1921, entirely new machines were designed, taking into consideration the ruggedness required in the electrical equipment, the overloads to which the motors were subjected and the higher cable speed demanded by the operators, which necessitated larger motors than previously made.

The motors used on present-day logging engines are specially designed machines having a very heavy overload capacity that gives them a maximum torque equivalent to a steam plant used for the same duty.

The motors are direct-connected to the drum systems by steel gearing, each motor having three shaft bearings—two in the motor housing and one outboard.

There have been five types of machines developed up to the present time, namely,

Two-speed yarders using a mechanical-change two-speed device
Two-speed yarders using a two-speed motor
Boom loaders using two motors, one loading and one swinging
Duplex loaders using two motors, one motor on each loading drum
Interlocking-drum-type skidders.

Reversing motors are used in most cases. The loading motors have the same number of control points reverse as in go-ahead, while the yarder motors usually have only one or two points of control reverse.

There are numerous ways of mounting these engines for use in the woods. A yarder and loader may be mounted permanently on a steel car which is always used at the railroad and moved by the locomotives, or a yarder and loader may be mounted separately on large timber sleds. These machines mounted are used away from the railroad, and in moving drag themselves by their own power through the woods. The skidders are always mounted on steel cars.

The general scheme of power transmission for electric logging is as follows:

In cases where power is purchased by the logging company from a power company, the latter usually installs a central substation on the logging company's property where the energy is stepped down to about 13,000 volts. The transmission lines are run from the substation to the donkey transformer where the energy is further stepped down to 550 volts, and it is then transmitted to the donkey control through armored cable which lies on the ground. Each electric machine has its own individual transformer.

The motors all have magnetic control with overload and undervoltage protection. A reversing knife switch is also provided be-

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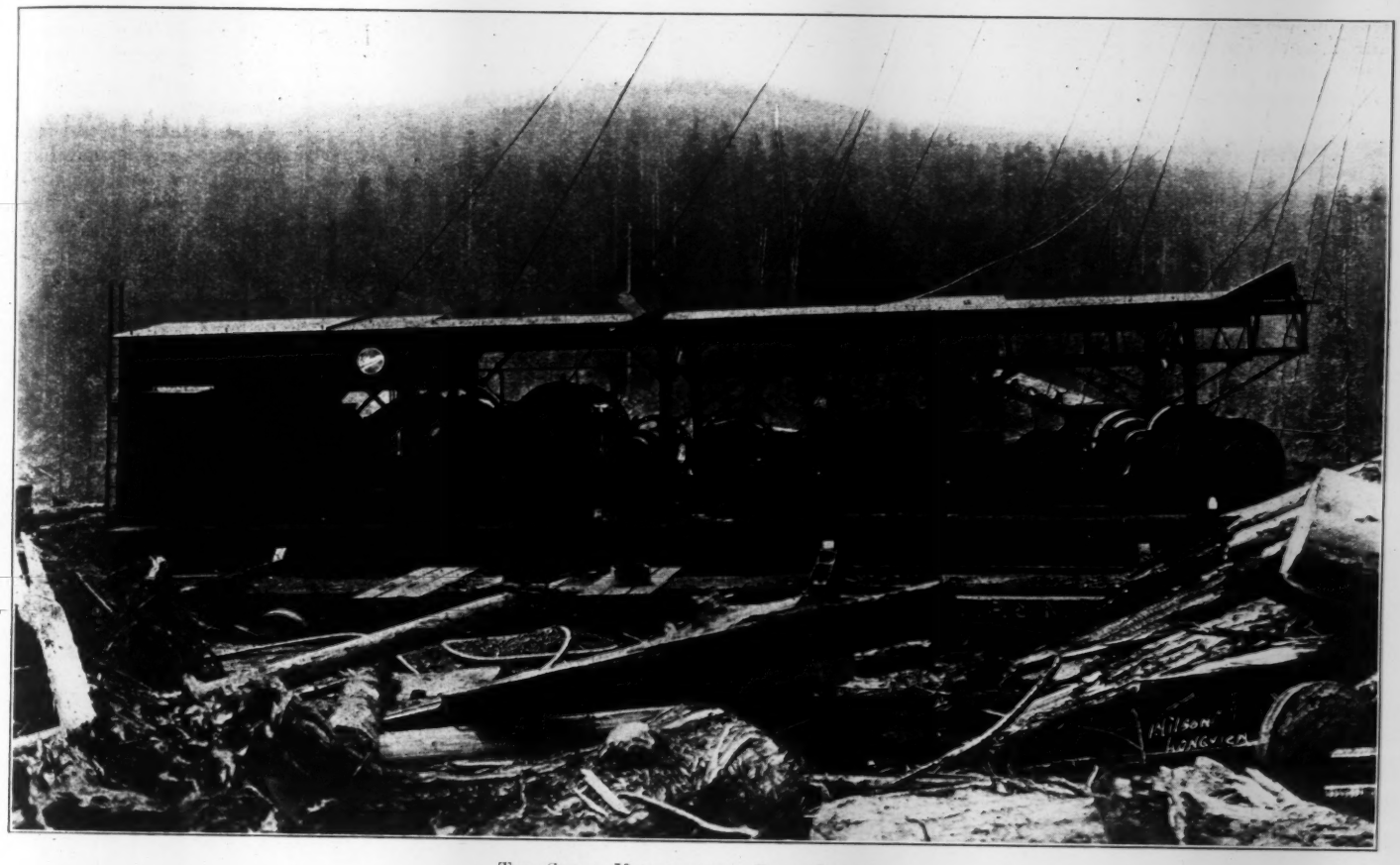
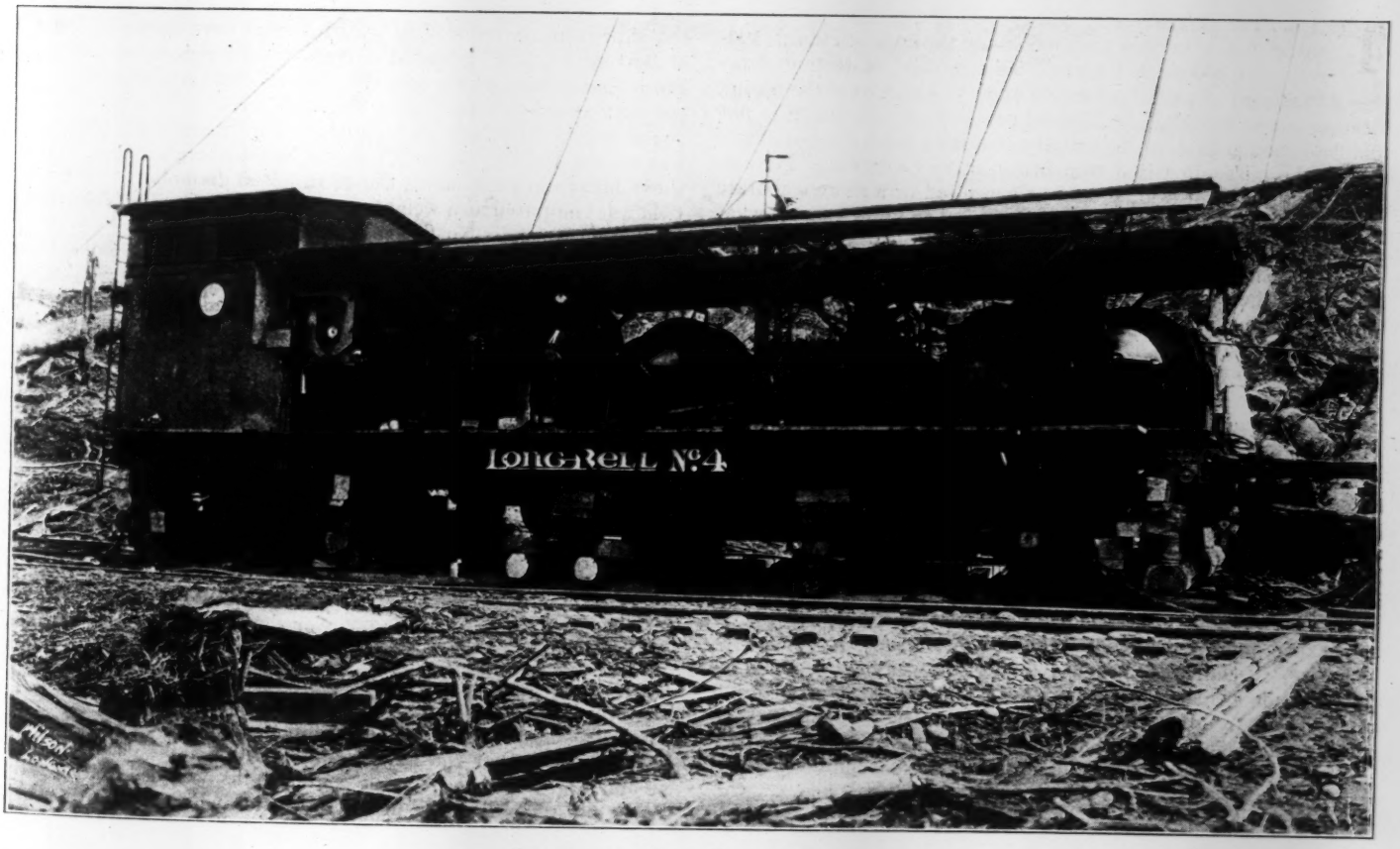
[1] Chief Engineer, Willamette Iron & Steel Works. Mem. A.S.M.E.

[2] A logging term meaning a combination of machine units and equipment that yards and loads logs.

Presented at the Portland Regional Meeting of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, Portland, Ore., June 22–25, 1925.



Two-Speed Yarder with Boom Loader



Interlocking-Drum-Type Skidder

tween the transformer and control panels for use in event of the wires being reversed in hooking up on a new logging setting.

The motors which do not act as brakes on these machines are either equipped with solenoid or air-operated brakes on the motor shaft. In both cases the brakes are operated through the control circuit.

In cases where the logging company generates its own energy at its mill, the energy is transmitted from the source of supply to a central switching station and from then on as before.

There are numerous advantages in logging by electricity, the most important of which are as follows:

1 Fire hazard eliminated so far as the logging engine is concerned

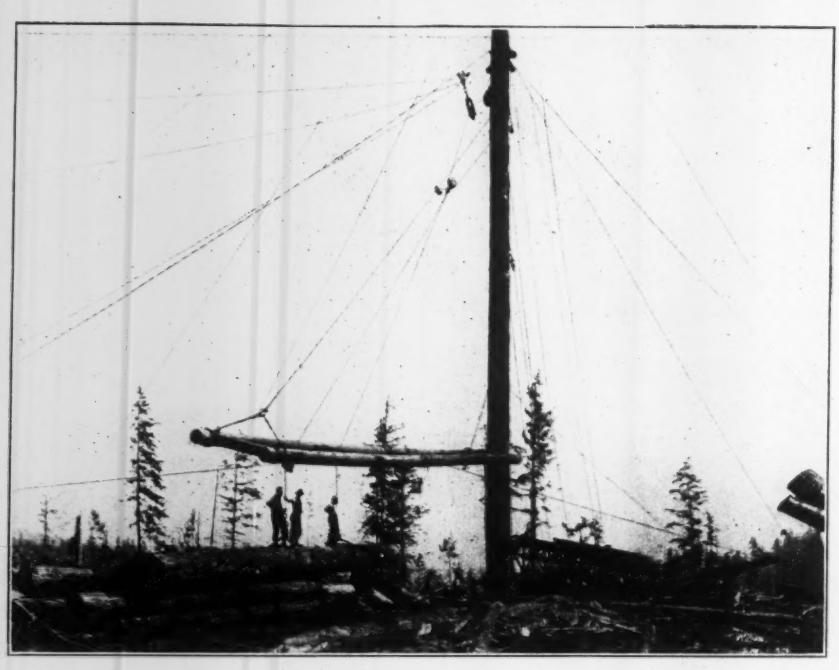
2 Fuel and water problems eliminated, which are quite costly in some parts of the country.

3 Fewer shutdowns than with steam, consequently more continuous operation.

The relative speeds of logging are somewhat higher on electric machines than on steam for the reason that the speed is better sustained on account of constant power. The steam machine is apt to have a drop in steam pressure, which necessarily affects the speed; this is more pronounced on long hauls.

It is very difficult to obtain accurate data regarding costs per 1000 ft. B.M. for electric energy due to the various contracts and operating arrangements each logging company has.

The energy consumed by a yarder and loader unit of present-day design in fir operation is approximately 10 kw-hr. per 1000 ft. of lumber logged.



Loading Boom and Spar Tree

Testing the Fineness of Pulverized Coal

THE fineness of pulverized coal is usually expressed as the percentages of a sample that will pass through standard 100-mesh and 200-mesh screens. This method of testing, however, according to a paper[1] presented by L. V. Andrews at the Spring Meeting of the A.S.M.E. in Milwaukee, is at fault in that it is not complete enough to show the actual pulverizing work done and may actually be misleading in a comparison of two samples. The microscope, the author said, should be consulted or a more elaborate screen test run in all cases where it is desired to know exactly what quality of coal is being delivered by the pulverizing mill.

In discussing Mr. Andrews' paper, W. J. Wohlenberg,[2] wrote that the size of the pulverized-coal particle played an important part, not only in the combustion process, but as well in the radiating powers of the flame. The author's study might thus be taken as an important step toward the accumulation of more specific knowledge concerning the physical conditions actually existing in the pulverized-fuel furnace.

The photomicrographs in the paper apparently indicated that the particles appeared in all manner of shapes and also that both fuel type and method of pulverization had an important influence in the distribution of the fines between the coarser and extremely small particles. In the light of some of the material presented by Professor Wohlenberg in another paper[3] at the same session, this was an extremely important disclosure.

It was unfortunate that the photomicrographs did not lend themselves for a more specific determination of the particle distribution as to size. A glance at the figures made it quite obvious that a much greater scale of magnification would be required to serve such purposes. Perhaps a large-scale enlargement of the photomicrographs might yield some additional information of interest.

The method suggested by the author of estimating fines by plotting curves had been utilized in making out the data contained in the paper already referred to. Such curves were shown in Fig. 22 of that paper, and indicated a large weight percentage of fines. For instance, with 85 per cent through a 200-mesh screen, about 50 per cent would pass through a 300-mesh screen. The relation between screen mesh and particle size as to mean initial diameter was shown by Fig. 23 of Professor Wohlenberg's paper. With these data from such curves the mean initial diameter d of the particle was computed as a weighted average with the results shown in Table 3 of Appendix No. 6 of that paper.

H. W. Brooks,[1] wrote that those familiar with the committee work and research effort of the American Society for Testing Materials, American Institute of Mining and Metallurgical Engineers, National Research Council, Bureau of Mines, and Bureau of Standards of the past few years to fix upon standard methods of sampling and testing finely pulverized materials, would readily agree with the author's conclusion that the standard 100- and 200-mesh percentage method of testing pulverized coal was at fault in that it was not complete enough to show the actual pulverizing work done, and that it might actually be misleading in a comparison of two samples. Equally, however, they would be led to wonder why the author himself utilized the method he condemned, even in his carefully qualified statement, to draw the conclusion that the performance of individual pulverizers could be compared to some extent by this method, and to advance in his tabulation, at least by inference, the quite astonishing and well-nigh revolutionary idea that certain pulverizers of the impact and attrition types might be expected to produce a higher percentage of impalpable fines than the mill classified as contact type. True, the paper did not definitely so state, but one could not fail to draw his own conclusions from the fact that the impact and attrition mills occupied first, second, and third places, while contact mills occupied fourth, fifth, sixth, and seventh places in the tabulation which the author had made.

The following statements in the paper would particularly astonish the coal photomicroscopist:

[1] A Microscopic Study of Pulverized Coal. Contributed by the Fuels Division and presented at the Spring Meeting of the A.S.M.E., Milwaukee, Wis., May 19, 1925. Published in Section 2 of Mechanical Engineering for May, p. 429.

[2] Associate Professor of Mechanical Engineering, Sheffield Scientific School, Yale University, New Haven, Conn. Assoc.-Mem. A.S.M.E.

[3] Radiation in the Pulverized-Fuel Furnace, by W. J. Wohlenberg and D. G. Morrow. See p. 627 of the August issue.

[1] Consulting Engineer, Fuller-Lehigh Co., Lehigh Co., Fullerton, Pa. Mem. A.S.M.E.

1 The samples obtained from a few mills were too small to admit of accurate screen testing but the manufacturers obligingly furnished the figures as listed.

2 Fifty magnifications were used to show the proportionate quantity of extreme fines to that of the coarsest in the sample being examined.

3 Moisture content at the time of pulverizing was not taken into account.

4 The way in which different coals break down during the process of pulverization was not taken into account.

5 Samples of impact pulverized coal were taken across the discharge pipe with considerable variation in both the 100- and 200-mesh material indicated.

Discussing these statements in the order given, Mr. Brooks wrote that:

1 Realizing the difficulties of accurate screening—the Bureau of Standards screen certifications permitted 8 per cent tolerance in average opening and 60 per cent tolerance in maximum opening on 200-mesh screens—the widely varying results obtainable with varying manipulations of hand and machine screening techniques, differences in end points, differing losses in weights due to sizing operations, and differing methods of cleaning sieves by tapping sides, brushing or rubbing with the fingers, one wondered as to the accuracy and consistency of the author's results and as to whether the impact- and attrition-mill samples which stood one, two, and three in the tabulation might not have been numbered among those the data upon which had been furnished by the manufacturer.

2 Bureau of Mines weight calculations[1] with pulverized coal magnified to 200 diameters showed variations between the calculated weight of coal on the slide and the actual weight ranging from a minimum of 1.19 to a maximum of 1.80. Another Bureau of Mines investigator[2] had said of photomicrographs (250 to 500 magnifications) that it was practically impossible to obtain one field of photographing that would be truly representative of the distribution of the sizes in a total sample. What order of accuracy or consistency of results the author expected, therefore, with a visual inspection of photomicrographs of 50 magnifications, was not apparent.

3 Any pulverizing method in any way employing air elutriation for separation would always be radically affected by the moisture content of the coal pulverized. Not only did it affect the power for pulverizing but the size of product as well in many mills. The mechanical energy of the driving motor, turbine, or engine was converted in the mill, through the tearing, rubbing, or grinding action of the pulverizing coal, and through friction, into heat, which vaporized whatever moisture might be present. Due to varying viscosities and surface tensions, it obviously made a vast difference whether the product was elutriated by dry air or steam.

4 Ten principal and hundreds of subsidiary plant products of as many different plants as there were in our forests today falling in all directions into varying stage of decomposition, putrefaction, decay, and moldering had contributed to the coal substance. Was it reasonable to expect the way in which different coals broke down during pulverization to be uniform? Coal consisting largely of anthraxylon (woody matter) from the bottom of a given seam would pulverize entirely differently and give entirely different impalpable dust content from attritus (ground mass) from the upper portion of the same seam. What chance was there of consistent results in a comparison where not even the same coals had been pulverized?

5 With the wide variations of fineness cited by the author in 100- and 200-mesh materials across the discharge pipe from the impact mill, what might one expect as to impalpable fines of 230-, 270-, and 325-mesh (4.4μ) on down to 1μ which investigators ordinarily sought with the microscope. These materials not only passed very close to the inner pipe wall, the coarser particles passing through the center, as the author stated, but likewise their velocities varied greatly as compared with that of the coarser materials. Hence unless the sampling pipe were held in its various position for time intervals proportionate to such velocities (the computation of which would be very difficult) and the samples aspirated at exactly these velocities, conditions of turbulent flow would prevail around the sampler orifice which might produce almost any results desired. In any case, the presence of the sampler created artificial flow conditions tending to throw the fines into suspension just at the point where the sampling was being attempted. In bin sampling in the cement industry, those versed in the subject claimed the ability to produce samples of any desired average fineness if permitted to select the sample themselves.

Henry B. Jones,[1] wrote that the distinction the author had made between "contact mills" and "roll mills, no contact" was not entirely clear, in that roller mills of the so-called contact type had practically the identical action described under the other heading. It was not intended in these roller mills that there should be any contact between the roller and the ring, nor was contact intended in the case of the Fuller mill, although in both of these mills contact did occur when the coal feed was momentarily light, just as it would occur in the mills classified as "roll mills, no contact."

The Fuller mill introduced a rubbing action between some of the grinding surfaces, being to this extent an "attrition mill."

The tube and ball mills had a distinct difference of action, depending upon the speed at which they were operated. Some of these mills were designed so that the balls were rolling on each other, whereas others were so designed that the balls were lifted and thrown or "cascaded." Roll mills employed only the rolling action, in which case the grinding effect was probably a combination of rolling and attrition. In those mills where the balls were lifted and cascaded there was a combination of rolling, attrition, and impact.

It must be conceded that there was much of interest and probable value to be learned from a determination of the character of those particles in a pulverized-coal sample which passed through a 200-mesh screen. It must also be conceded that screening tests of these finer particles were very unsatisfactory and difficult to make. Screens with 300 and 350 meshes to the inch had been made and used, although there was more or less question as to the accuracy of results so obtained, and there was still much undetermined regarding the character of this fine material beyond the range of these screens.

Microscopic studies had been made at various times in the history of the development of pulverized coal, but without ever producing results of a definitely satisfactory character. The chief difficulty in such work was to find a method which would "hold the particles in suspension and prevent fines from accumulating in masses which resembled large particles." Those who had undertaken work of this character would hardly be willing to accept results of such studies unless they were well satisfied that the medium so used had fully accomplished its purpose and prevented any contact between separate particles. Obviously it was very difficult to establish this point.

The author also referred to the necessity for the careful selection of a field of view which would show a uniform density of particle distribution in various photographs. It would appear that such a selection was comparable to selecting samples which would give desired results. Certainly a method of this sort could not be relied upon to serve as a mathematical analysis of the sizing of various samples.

Quite probably the method of plotting a curve from screen sizes above 200 mesh and extending it to secure an indication below 200 mesh was fully as effective and accurate a method of determining the fines distribution as the microscopic study and it certainly was much simpler of accomplishment.

The author was quite right in his statement that "the method of sampling determines to quite an extent the accuracy of the samples collected" and he was quite right in emphasizing the necessity for care in sampling the output of unit pulverizers.

Mr. Jones also agreed with the author's conclusions that the standard 100- and 200-mesh percentage method was not complete enough and that more elaborate screen tests were desirable, and recognized that the microscope was a valuable tool, which might, in the hands of an experienced and careful operator, throw more or less light upon the character of the fines in a given sample.

John Anderson,[2] who acted as chairman of the session, said, in opening the oral discussion of the paper, that his experience had

[1] G. St. J. Perrott and S. P. Kinney, The Meaning and Microscopic Measurement of Average Particle Size, *Journal American Ceramic Society*, vol. 6, no. 2, Feb., 1923, p. 436.

[2] W. M. Weigel, Size and Character of Grains of Non-Metallic Mineral Fillers, B. of M. Technical Paper 296, p. 37.

[1] Fuller Lehigh Co., New York, N. Y. Mem. A.S.M.E.

[2] Chief Engineer of Power Plants, Milwaukee Electric Railway & Light Co., Milwaukee, Wis. Mem. A.S.M.E.

showed that the matter of fines in the smaller grades was not so important as the coarser grades. In a number of tests he had found that as low a percentage as 58 through 200 mesh did not interfere with combustion. However, a furnace that would burn in suspension coal of 30 mesh had not been built. In the future, he said, he looked for a combination furnace in which the fine particles could be burned in suspension while the coarser would drop to the bottom and burn on a grate.

He said he believed that the whirling streams of flame in the well-type furnace described by H. W. Brooks in his discussion of the Christie paper, would greatly reduce furnace volumes.

In closing the discussion the author said that Mr. Brooks had attempted to use the information contained in the paper for purpose of mill comparison. Such was not intended at the time the investigation was started. The purpose of the investigation was simply to prove or disprove whether or not the standard of 100- and 200-mesh screens could be depended upon to show the nature of extremely fine coal.

The classification of mills as given in the paper, he said, was simply to show that there was a reasonable possibility of a dissimilar breaking down of the coal due to the mill itself. No attempt was made to be extremely explicit nor to identify any one particular type of mill.

The author said he was gratified to learn that investigations of a similar nature were being conducted by the Bureau of Mines and others. If it should be later proven that the screen test ought to be amplified, or that finer screens should be used, this would be brought out.

The Modern City and the Engineer's Relation to It

By ROBERT RIDGEWAY,[1] NEW YORK, N. Y.

IN A MATERIAL sense, the modern city is the result of the work which has been done by the pure scientist and the engineer. Without them the city could not exist. To their skill and toil are due the water supply, the sewers, and the sanitary conveniences, the paved streets, the transportation facilities for local, suburban, and interurban use, the telephones, the electric lights, and all the many other facilities on which a city has come to depend for its very existence. Nor could the city be kept free from epidemics and its health be maintained at so high a standard without the excellent work of the sanitary engineer in coöperation with the chemist, the bacteriologist, and with medical research. So it has come about that typhoid and yellow fever have been largely eliminated and nearly all communicable diseases have been brought under control.

The change to urban life has affected deeply the customs, the habits, and the thoughts of the people. In the simple days during the early life of the Republic, when men lived generally distant from one another, they were largely individualists, believing with Thomas Jefferson that government was a necessary evil and that they should get along with as little of it as possible, depending on their own efforts for success. Modern concentration of vast numbers of people in cities has led to the organization of industrial corporations with hundreds and frequently thousands of employees working under the same roof. Diversities of custom are disappearing. Organizations based on class consciousness are formed with the idea of bettering the material condition of particular classes of workers. More government is demanded. There is a drift toward paternalism and socialism; a tendency to lean on the state or national government for help out of all difficulties; to lose that Anglo-Saxon spirit of independence which formerly prevailed and which was the foundation on which the structure of our national life was built. The fear exists that these tendencies are going too far; that the worker is being made into a machine and that so much effort is given to developing his material side, that the moral and the spiritual sides are forgotten. While city workers have escaped the hardness of the farm life of the old days, they are paying for the ease of life in a loss of that touch of fundamental things which is so necessary for true happiness and for the full rounding out and proper balance of humanity.

The price we have paid for the many advantages which life in the great cities has given us is a large one. The opportunities and advantages of urban life have not yet come to compensate humanity for the restful quiet of the open country, for the simple pleasures it affords, and for the spirit of introspection which it fosters. Their comforts and luxuries are to be enjoyed only at the expense of a certain softening of character. The joy of doing is marred by the prevalent feeling of unrest, and the ever-mounting cost of living in cities is a cause of anxiety to those of moderate means. It is becoming increasingly expensive to bring foodstuffs and other supplies into large cities, principally because of the restricted terminal facilities, which are, however, in part, the result of a narrow or selfish vision of the particular community.

Large cities have come to stay and will doubtless continue to exist as long as industrial conditions and human nature remain as they are and as long as gregarious man retains the desire to live and work where other men congregate. Recognizing this, the efforts of every good citizen should be directed to making cities what they should be, by each giving of his best thought to improving the quality rather than the number of inhabitants. Many are giving thought to the proposition that it would be better for the well-being of the human race if the tendency it now displays toward urban life could be checked. It is a grave question whether the future rapid growth of cities should be encouraged. It is obvious that our cities cannot continue to grow until they include the entire population. There must come a time when their growth will be checked by the inevitable process of economic laws, if thinking men do not find some way to check it before that time arrives. It does not conduce to the well-being of the people to have cities become too large. It is not economically sound to have them do so. The cost of transporting the people workward and homeward, of bringing food to them, and of providing for them in other ways, adds a great load to an already burdensome tax not only on themselves but on the other people of the nation.

I have referred to the part the engineer has played in the upbuilding of the modern city and of the debt the public owes him, but, if he has done all these wonderful things, he has assumed a responsibility for them and owes a debt in turn. We cannot bring something into being without assuming responsibility for its proper development and for the use that is made of it. After all, works of the engineer's creation are but the means to an end, not the end itself. They contribute to the well-being of humanity and man is more than a physical being. Surely, the engineer has a duty to perform in addition to the development and care of material things, wonderful as they are. I do not like to think that the engineer can see only the steel and masonry structure he designs and builds. I hope he has a vision of what it is built for as the architect of the Middle Ages had of the great cathedral he erected, and that the spiritual, moral, and esthetic sides of life have a great value to him.

Thanks to the widom of our forefathers, the foundations on which our country's institutions rest are broad and deep. Incorporated in the design where high ideals and the superstructure was raised in fidelity and in faith. The task of its maintenance is that of each succeeding generation. As in the case of all structures built by man, repairs, renewals, and additions are necessary, but do not call for destruction of the original fabric. The skeleton of the structure must be maintained intact. The larger duty of the technical man calls him to this task. To the task he brings special qualifications. As a practical idealist his patriotism and civic spirit should be manifested not by a waving of flags and by boasts of superiority to others, but by searching out the defects in our present make-up and then by aiding others to raise the standards of civic affairs in the same deliberate and constructive way in which he designs and builds his physical structures for the use and benefit of mankind.

[1] President, A.S.C.E. and Chief Engineer, Board of Transportation, City of New York.

Excerpts from the Presidential Address at the Annual Convention, Cincinnati, Ohio, April 22, 1925, of the American Society of Civil Engineers.

Mechanical Features Affecting the Reliable and Economical Operation of Hydroelectric Plants

By E. A. DOW,[1] WORCESTER, MASS.

This paper deals with the subjects of flashboards, trashracks, rack rakes, sluice and headgates, penstock valves, air vents, and discharge-measuring devices from the standpoint of economy and reliability of operation, based almost entirely on the New England Power Company's practice. It stresses the importance of reducing head losses and leakage and adhering to strength and simplicity in the design of hydraulic equipment.

THIS paper presents a series of comments and opinions on some of the mechanical features affecting the reliable and economical operation of hydroelectric plants. By far the greater part of the author's experience has been with the plants of the New England Power Company, and a brief description of this system will be helpful to an understanding of the matters discussed.

The New England Power System

As will be evident from Fig. 1, the New England Power Company serves a highly developed industrial territory. It owns seven hydro plants, of an aggregate capacity of 115,000 kw., as well as a small steam station. In 1924 the system output was 563,000,000 kw-hr. and the maximum peak load was slightly over 150,000 kw.

At the Vernon Station the installation consists of eight 2500-kw. vertical three-runner units under 34 ft. head, and two modern single-runner 5000-kw. units. Three plants on the Deerfield River at Shelburne Falls, Mass., each have three double-runner horizontal units of 2000 kw. capacity. The heads on these plants are 60, 64 and 66 ft. The Somerset Reservoir at the headwaters of the Deerfield is intended for storage, consists of a 110-ft. earth dam impounding 2,500,000,000 cu. ft. of water. The plant known as No. 5, about three miles above Hoosac Tunnel, consists of three single-runner horizontal units set in cast-iron scrolls and operating under 240 ft. head and has a capacity of 17,000 kw. A plant on the Deerfield at Searsburg is a single-unit installation having a capacity of 5000 kw. under 230 ft. head. It is a completely automatic plant which has operated with entire success in spite of hydraulic complications such as 18,000 ft. of wood-stave conduit, differential surge tank, etc. The latest addition to the Deerfield list is the Davis Bridge Plant near Readsboro, Vt. This is primarily intended for storage and has an earth dam 200 ft. high forming a reservoir of 5,000,000,000 cu. ft. capacity, but it is also the largest power plant on the system, with three 15,000-kw. units operating under a head which varies between 300 and 390 ft.

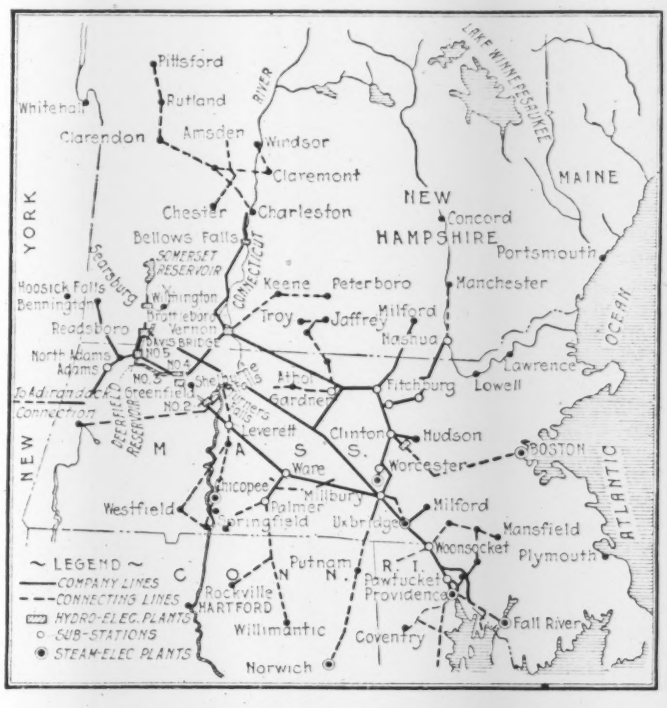


FIG. 1 THE NEW ENGLAND POWER SYSTEM

Flashboards, Crest Gates, etc.

The ideal crest-control device would be entirely automatic in operation and would hold the water elevation above the dam between definite and comparatively narrow upper and lower limits; it would be tight, unaffected by ice or other interferences, leave an unobstructed crest under flood conditions or at least present openings of sufficient size to obviate danger of clogging, and be reasonable in cost.

The devices actually in general use include simple flashboards supported by pipes or pins designed to bend over and clear the crest when the water surface reaches a predetermined elevation, rather elaborate arrangements balanced hydraulically or by counterweights and dropping or rising automatically with a change in water level, and crest gates of the sector or sliding type which depend entirely on the human element for their operation. Probably no available type combines all the desirable features, and for each installation a compromise choice must be made.

Excepting those of the storage developments, all the spillways of the New England Power System are equipped with the simplest type of flashboards, ranging in height from 4 to 10 ft. The boards are supported by steel pipes and failure takes place with very satisfactory consistency at the desired pond elevations. When there is sufficient warning of an impending flood, the boards are removed, otherwise they are lost, which occurs probably once a year. Removal and replacement of boards is effected by means of boats and cableways. All plants are provided with sluicegate capacity proportioned to the drainage area, so that after the peak of a flood has passed the pond can be sufficiently lowered to permit of replacing boards. Satisfactory tightness is maintained by calking with cinders, the flashboards being assembled in sections, which aids in this respect. The system is preferable to any other yet devised. It is low in first cost, and more reliable than any device depending entirely on human vigilance. Among dangers to be guarded against are a temptation to brace pins to carry higher boards during months when the chance of sudden floods seems remote, and the tendency toward the formation of ice on the downstream face of boards and the upper part of the dam crest which may reinforce the pins so that failure will not occur until the pond level rises materially above the desired elevation.

At Somerset and Davis Bridge conditions affecting flashboards are materially different from those at other plants. The large storage capacities in the upper levels of the reservoirs preclude a dangerously sudden rise in water elevation, and on account of the great value of the stored water it is out of the question to draw down materially to replace flashboards after a flood. A control is required which can be replaced with a considerable depth of flow over the crest, yet it must be of such a type that it can be removed with certainty, by one man if necessary, in spite of ice, etc. Because of the lack of velocity of approach to the spillway of a large reservoir, there is much less danger of spillway openings becoming clogged with floating trees, ice floes, etc., than with a river spillway, consequently the objection to piers on the crest is greatly reduced. The type of control which seemed best for both Somerset and Davis Bridge consists of stoplogs supported by needle beams which can be tripped from a bridge spanning the openings between spillway piers. Fig. 2 shows a general view of the arrangement at Davis Bridge. The spillway is circular in plan, 160 ft. in diameter, the discharge passing down a shaft $22\frac{1}{2}$ ft. in diameter and out through

[1] Mechanical Engineer, New England Power Construction Co.

Contributed by the Power Division and presented at the Spring Meeting, Milwaukee, Wis., May 18 to 21, 1925, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. Abridged.

FIG. 2 DAVIS BRIDGE SPILLWAY

a horizontal tunnel located about 170 ft. below the crest. There are 16 radial piers and the span between each pier is broken up by three needle beams supporting the stoplogs. The beams can be tripped from the bridge by means of the latch at the top, thus releasing the stoplogs. They are still held by the chains, however, and can be pulled up and replaced at any time, the tops being returned to the latch and the bottom ends allowed to drop into the sockets at the bottom.

SLUICE GATES AND ACCESSORIES

Either surface or submerged sluices can be used, but for handling fairly large flows the submerged type ordinarily proves more economical and generally attractive than the other. Discharge may take place directly into the air, which permits of convenient inspection for leakage, etc., or below tailwater, which is less subject to trouble from freezing. The commonest kind of submerged sluice consists simply of a cast-iron frame and sliding gate provided with a screw stem and mechanical hoist or with a pressure cylinder. The latter is very much to be preferred when the operating mechanism is located in a damp place, as a mechanical hoist, particularly if electrically operated, is difficult to maintain under such conditions.

Fig. 3 shows one of the seven 9- by 7-ft. Vernon submerged sluice gates which have an instructive history. The original gate installation at Vernon consisted of eight gates of standard design, controlled by bronze stems and electrically operated mechanical hoists. The gates were light, the tensile stress in the cast iron running as high as 4000 to 4500 lb. per sq. in. The hoist capacity had been figured on too low a friction coefficient and the hoists were overworked. The stems were correspondingly light and had probably been designed on the assumption that, being of bronze having the same tensile strength as steel, they would be equal to steel as columns, no attention being paid to the lower modulus of elasticity. Several of the stems buckled. Sections of bronze wearing strips on the gates and frames broke out, causing excessive leakage. The gates themselves gradually failed under the load. The entire set of gates and frames were finally replaced with much heavier equipment especially designed to meet the Vernon conditions, using tensile stresses under 1500 lb. per sq. in. in cast iron and substituting oil-pressure-operating cylinders for the mechanical hoists. Upon unwatering the discharge passages it was found that in several cases the cast-iron lining to which the gate frames bolted had been loosened up, cracked, and warped, probably from the effect of headwater pressure between the lining and the concrete combined with vibration. In connection with the replacement another source of trouble was eliminated by the removal of a grillage of horizontal bars originally installed to prevent stoppage of the openings by water-logged stumps, etc. This grillage did more harm than good by catching trees and logs after they had passed part way through and holding them where they would interfere most with the closure of the gates.

From the Vernon experience it was learned, among other things, that:

a It does not pay to economize on first cost by using light sluice-gate equipment.

b Designs should be based on a coefficient of sliding friction of at east 0.75.

c Bronze should not be used in compression when it can be avoided. Steel protected by a bronze sleeve is preferable.

d Metal liners below free discharge gates should be properly anchored back into the concrete.

e If a grillage is used before a sluice gate it should be located sufficiently far upstream so that debris caught in it will not trail down stream far enough to obstruct the gate.

Plain sliding sluice gates can be expected, if properly designed,

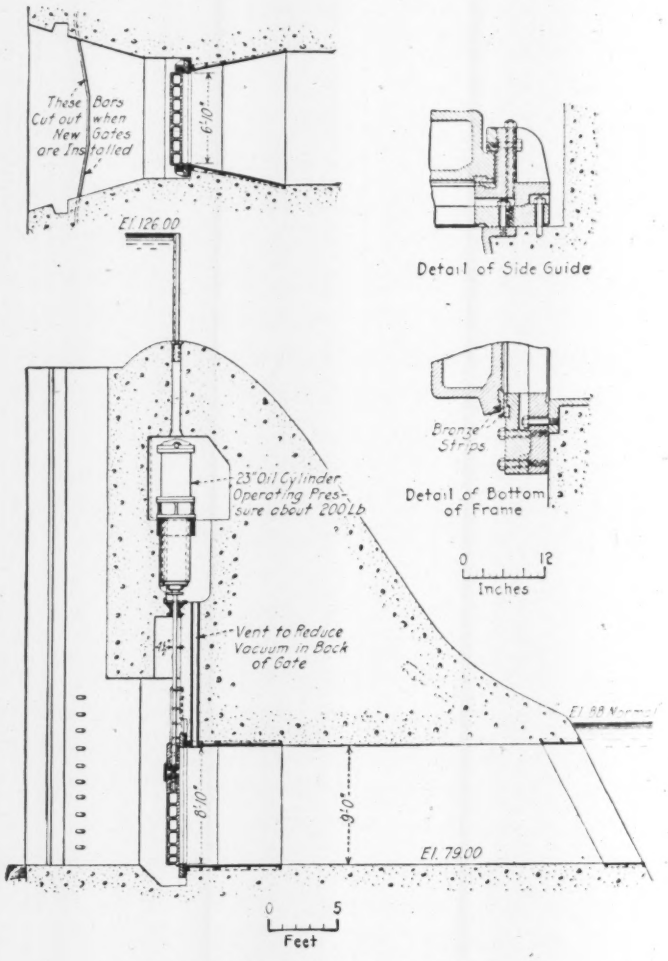


FIG. 3 VERNON SUBMERGED SLUICES

to operate satisfactorily for free discharge service up to heads of 50 to 75 ft. For higher heads the vibration is likely to be excessive and some other type becomes preferable. Several recent installations have made use of a modified design of butterfly valve, known as the disk-arm pivot valve.[1] This valve differs from the ordinary

[1] Patents pending.

butterfly in that it is controlled by a rod connecting with an arm cast integral with the disk rather than by a crank on the end of the spindle. This gives an extremely strong and rigid operating mechanism with no vibration under high velocities. Fig. 4 illustrates a setting where four 72-in. disk-arm valves are being installed under a head of only 37 ft.

It is a common but decidedly questionable practice in sluice design to flare the passage below the gate or valve to increase the discharge by reducing the velocity head lost at exit. Under low heads a considerable increase in discharge capacity can be obtained, but the percentage effect rapidly falls off as the head goes up. A very trifling flare generally suffices to reduce the pressure at the valve to absolute zero, and the most constricted part of the passage becomes the neck of the bottle. It is never possible to increase the discharge beyond that corresponding to the net head plus 34 ft. acting on this neck. Designs in which the pressures approach absolute zero result in a greatly increased tendency to vibration, and the combination of low pressure and high velocity is ideal for the pitting of metal and erosion of concrete. Vents admitting air to the discharge passages immediately below valves or gates serve to reduce the tendency to vibration and pitting somewhat, but are not a complete cure for the trouble. Increased maintenance expense due to the flare is likely to eat up quickly any saving in first cost made by the use of a smaller gate or valve.

Racks and Accessories

In most installations the water leaving the pond or reservoir passes through racks whose function is to screen out debris but to pass the required amount of water without excessive head loss. This seems like a simple requirement, yet clogged racks have probably been responsible for more serious hydro-plant shutdowns than any other one factor. The design of a suitable rack involves the size and type of turbine, the character of trash, the method of cleaning, allowable head loss, ice conditions, and any special local peculiarities. The rack area should be as generous and the velocities as low as possible, as difficulties increase rapidly with higher velocities. In general satisfactory results may be expected from velocities in the neighborhood of $1^1/_2$ ft. per sec. figured on the gross rack area. In New England frazil ice or a run of leaves is most likely to interfere with rack operation. The former is apt to clog racks completely, even with rather widely spaced rack bars, and all that can be done is to pull the racks and take a chance on trash which may enter the turbines. It is possible to cope with leaves by continuous raking, and if bars are spaced about $2^1/_2$ in. apart, most of the leaves pass through and cause little trouble.

It is possible to rake racks satisfactorily by hand to a depth of 15 to 20 ft., but with depths of 30 to 40 ft., such as are being found necessary in large low-head plants, hand raking is out of the question. Mechanical rakes are now available. In a design for use at the Vernon Station a traveling motor-operated rake can be moved to any desired bay and dropped down in suitable fixed guides with which the traveler is lined up. The rake is approximately 20 ft. wide and cleans an entire bay at one operation. During the spring high water the rack loss frequently amounted to several feet and under the worst conditions it was necessary to shut a unit down in order to rake it successfully. The annual loss in output at Vernon due to clogged racks has been estimated at about 3,000,000 kw-hr and the cost of hand raking has amounted to from $2500 to $3000 per year. The total annual loss therefore justifies expenditure for a device eliminating an appreciable percentage of it.

Headgates and Operating Mechanism

Headgates are in general protective devices used to shut off the water passages and turbines in case of accident or when drainage is necessary for inspection, repairs, etc., and so far as possible,

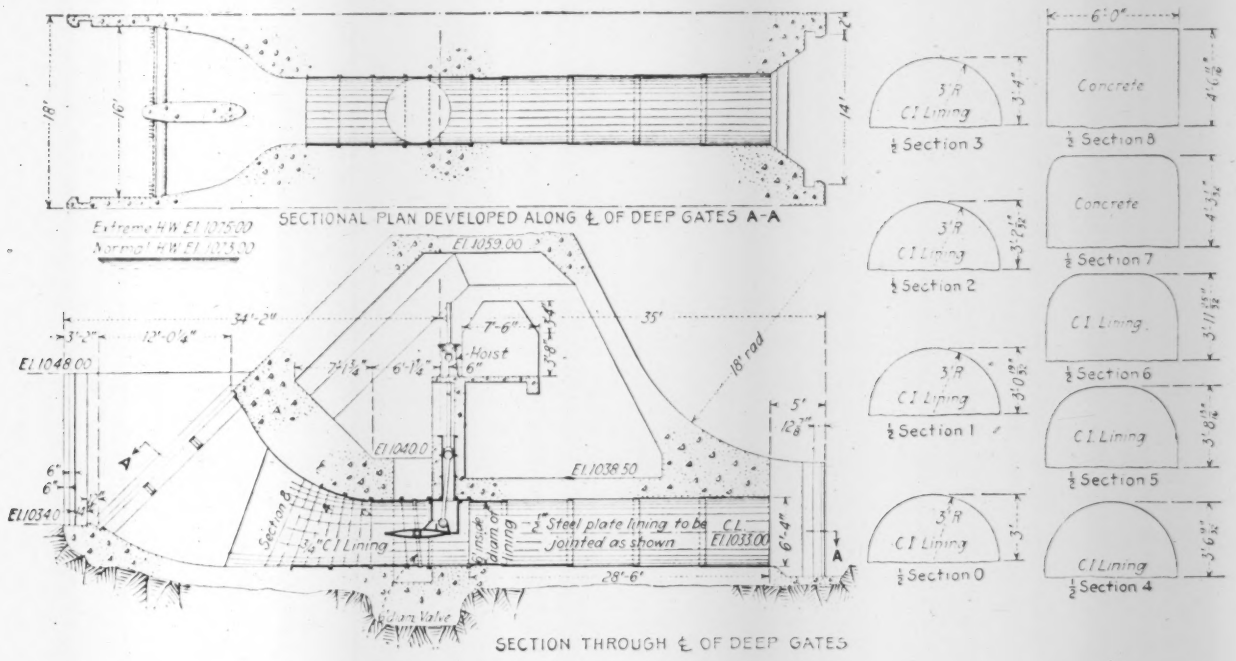


FIG. 4 BRASSUA LAKE VALVES

they should be able to close in an emergency in spite of the maximum discharge which could occur under any probable condition, be reliable and reasonably rapid in operation, sufficiently tight or so designed as to be easily calked so that leakage would not interfere with work to be done below them, be free from ice trouble, safe against accidental opening, foolproof, and either protected from obstructions or designed to close against them without damage. They should cause a negligible head loss and should be reasonable in first cost and maintenance.

Headgates operating under high or low pressure, while performing the same functions, are likely to vary widely in type. A fairly typical low-head installation is illustrated in Fig. 5, which shows the gates used for the Vernon extension units. Each of these units has a capacity of 5000 kw. under 34 ft. head and discharges about 1800 sec-ft. Each unit has two gates 17 ft. 6 in. high by 15 ft. 9 in. wide, operated by 28-in. oil cylinders. These gates are designed to close in an emergency under full head with no back pressure, so that it is possible to shut off the flow with the turbine gates open or with the turbine entirely wrecked. To aid in closing under emergency conditions the gates were made very heavy (120,000 lb. apiece) the construction consisting of a structural-steel frame filled with concrete. The weight of the gates plus the thrust of the hoists is sufficient to insure closure with a coefficient of friction of gate on guides of well over 0.75.

These gates are of the plain sliding type with seats of st

steel bearing on cast-iron frames. In common with most gates of this design they are not exceedingly tight, but the setting is such that except along the bottom edge they are easily calked by the use of cinders.

Cylinder hoists such as are used for these gates are extremely rugged and free from complication. A relief valve prevents the development of excessive pressure and nothing is injured if the gate closes against an obstruction, as the pressure simply builds up until the valve discharges. When used with heavy gates, however, they involve some type of latch to sustain the weight when the gates are raised, and this is a complication if it is desirable to provide for closure by remote control in an emergency. The Vernon gates are supported when open by struts of heavy pipe and it is necessary to raise them and release these latches before closure. If the cylinders are used in a cold climate in an exposed position they must be operated with some liquid such as hydraulic-jack oil which will neither freeze nor become so viscous as to refuse

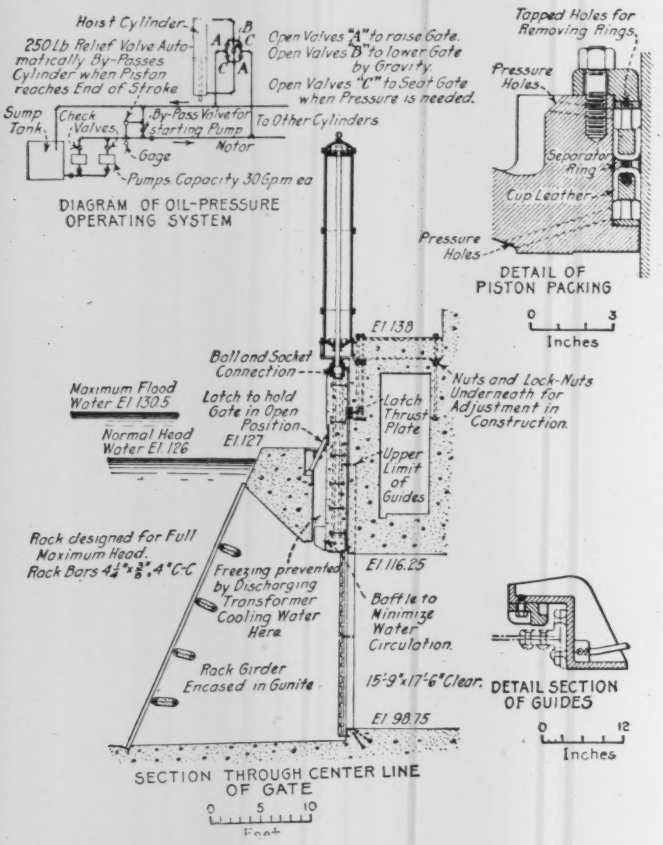

Hoist Cylinder
250 Lb Relief Valve Automatically By-Passes Cylinder when Piston reaches End of Stroke
Open Valves "A" to raise Gate.
Open Valves "B" to lower Gate by Gravity.
Open Valves "C" to Seat Gate when Pressure is needed.
Sump Tank
Check Valves
By-Pass Valve for starting Pump
To Other Cylinders
Motor
Gage
Pumps Capacity 30 Gpm ea
DIAGRAM OF OIL-PRESSURE OPERATING SYSTEM
Tapped Holes for Removing Rings
Pressure Holes
Separator Ring
Cup Leather
Pressure Holes
DETAIL OF PISTON PACKING
0 3 Inches
El 138
Ball and Socket Connection
Maximum Flood Water El 130.5
Latch to hold Gate in Open Position El 127
Normal Head Water El 126
Latch Thrust Plate
Nuts and Lock-Nuts Underneath for Adjustment in Construction
Upper Limit of Guides
Rock designed for Full Maximum Head. Rock Bars 4¼"x3⅜", 4'C-C
Freezing prevented by Discharging Transformer Cooling Water Here
El 116.25
Baffle to Minimize Water Circulation
Rock Girder Encased in Gunite
15'-9"x17'-6" Clear.
DETAIL SECTION OF GUIDES
0 12 Inches
El 98.75
SECTION THROUGH CENTER LINE OF GATE
0 5 10 Feet

FIG. 5 VERNON EXTENSION HEADGATES

to flow. Stems and cylinder walls must be smooth, otherwise the wear on the packing and consequent loss of oil will be excessive. Ample pump capacity is necessary so that cylinders can be operated in spite of minor leakage past the piston, and the New England Power Company has adopted a minimum pump capacity of 30 gal. per min. for such installations for this reason.

The lowering of gates of this type is ordinarily accomplished by bypassing the operating fluid from the bottom to the top of the cylinder, the weight of the gate pulling on the piston developing the pressure which forces the fluid from the bottom to the top. This means that motion is taking place *against* the pressure, and unless the piston leathers are provided with a separator ring of exactly the correct shape there is a tendency for the lower cup or U-leather to be drawn upward past the upper leather. The result is a severe folding and wedging against the cylinder walls, and the leather soon wears through. A further peculiarity becomes evident if pressure is admitted to the top of a cylinder while a heavy gate is being lowered or is being sustained by the pressure below the piston. Under this condition the pressure from the pump is superimposed on that developed by the weight of the gate and the result is quite likely to be a pressure approaching twice that which the designer anticipated. This possibility should be avoided by the proper arrangement of connections or by relief valves. Most of the troubles experienced with hydraulic cylinders are vexatious rather than disastrous and are quickly rectified, while a wrecked mechanical hoist is a very serious matter.

The Vernon headgates probably meet the requirements of the situation more completely than most similar installations. Men can work behind them without interference from leakage or involuntary opening. Interference with closure due to freezing is entirely eliminated by piping the slightly warm discharge from the water-cooled transformers into the gate wells, which are planked over to better retain the heat.

An interesting intake using the so-called "caterpillar" gate made under the Broome patent is illustrated in Fig. 6. This shows the arrangement used at the Rainbow plant of the Farmington River Power Company. The Broome gate is built with a seat inclined to the guides in such a way that the water pressure on the upstream side of the gate has a component in the direction of closure, so that the gate is self-closing if the friction is sufficiently low. In the caterpillar design frictional resistance is largely eliminated by the use of an endless roller chain and a hoist of the tension type only is required, since it is not necessary to force the gate down in closing. The arrangement originally considered for the Rainbow plant called for two caterpillar gates each 14 ft. high by 10 ft. wide for each unit. Further study indicated, however, that the cost could be greatly reduced by the use of one sliding gate 14 ft. high by 12 ft. wide and one caterpillar gate 14 ft. high by 8 ft. wide. The combined area is the same and the sliding gate will be weighted with concrete so that it will be heavy enough to close by its own weight under full turbine discharge, with the caterpillar gate open. When the sliding gate is down the caterpillar gate will be closed under full head, and of course the latter will be opened first when it is desired to fill the scroll. This installation cost about two-thirds as much as the design using the two caterpillar gates and is equally good.

The intake from the Davis Bridge reservoir in which the gates operate under relatively high head is shown in Fig. 7. The water level in this reservoir fluctuates over 90 ft. and it was of course necessary to locate the gates below the minimum water elevation. Actually, they are subjected to a maximum head of about 110 ft. The intake design adopted consists of a dry concrete tower 117 ft. high having an inside diameter of 20 ft., in the bottom of which are located two 8-ft-diameter valves. These valves are motor-operated and are of the disk-arm type previously referred to. They were adopted in preference to rectangular sliding gates which were less attractive because of the excessive hoist capacity required for reliable operation under the existing head, difficulties to be expected from damaged seats under the heavy pressures, etc. In butterfly valves the danger of cutting seats is minimized, since no sliding takes place on them, and in the design adopted the seats are of babbitt, which is very resistant to erosion and can easily be repaired. In determining the hydraulic characteristics of pivot valves, it was found that under any considerable velocity they have a powerful tendency to close as soon as the disk is moved from the wide-open position. The construction adopted was preferable to the standard butterfly because the use of an arm cast integral with the disk, permitting direct connection of the operating mechanism, consisting of stem, connecting rod, and crosshead, facilitated the development of a design insuring reliability of closure even under free spouting conditions, such as with a blowout in the tunnel. In this installation the disk-arm type was attractive for the additional reason that the central hoist arrangement made possible a closer spacing and consequently a smaller tower than could have been used had the valves been controlled from the ends of the spindles.

These valves have proved extremely tight, probably fully as tight as sliding gates, and certainly much more so than the standard butterfly. On several different closures one of them was found to show a leakage too small to measure, while the other leaked between 5 and 6 gal. per min. under 100 ft. head. This unusual tightness is attributable to the fact that the thrust of the hoist opposes the water pressure on the trailing half of the disk and prevents deflection, forcing the edge tightly against the seat around the whole circumference.

Davis Bridge is the largest single plant on the New England Power System, and to secure continuity of service every precau-

tion has been taken to insure against interruption and provision made for closing off either valve for repairs while the plant is supplied through the other by the use of upstream and downstream bulkheads which can be quickly installed when required.

To economize on size of tower and cost of valves fairly high velocities are used in the Davis Bridge intake design. At the maximum expected discharge of 1600 sec-ft. the actual velocity at the most constricted section reaches about 19 ft. per sec. The water passages were carefully designed, however, particularly in the flaring portions, and the actual head losses are gratifyingly low, the intake loss amounting to less than one-half of one per cent of the total head.

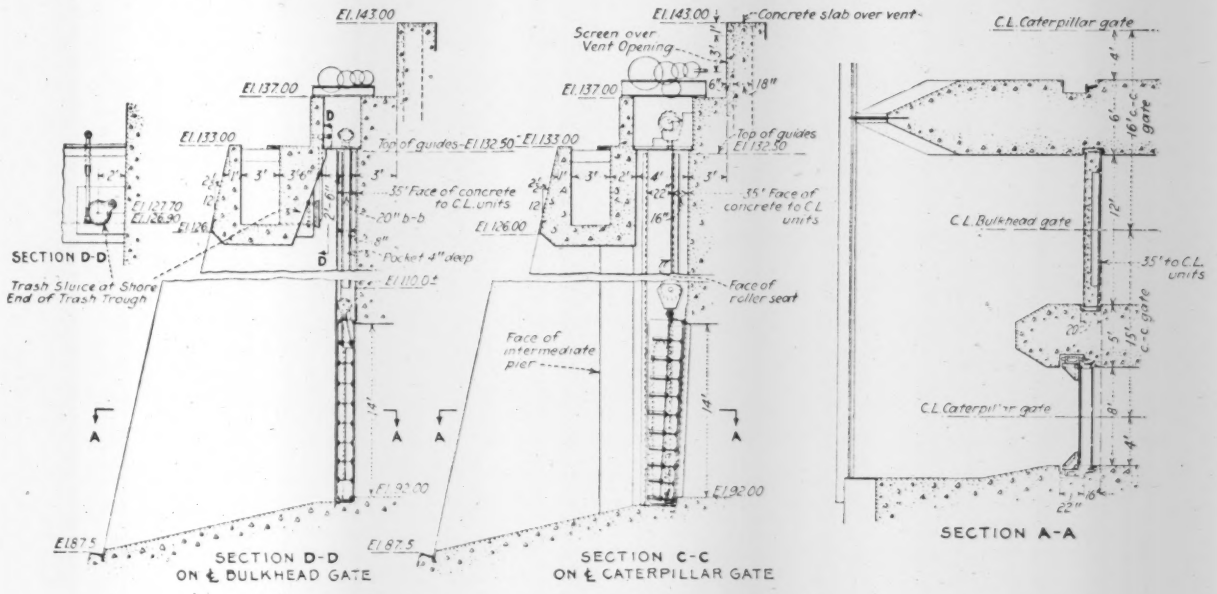


FIG. 6 INTAKE GATES, FARMINGTON RIVER POWER COMPANY

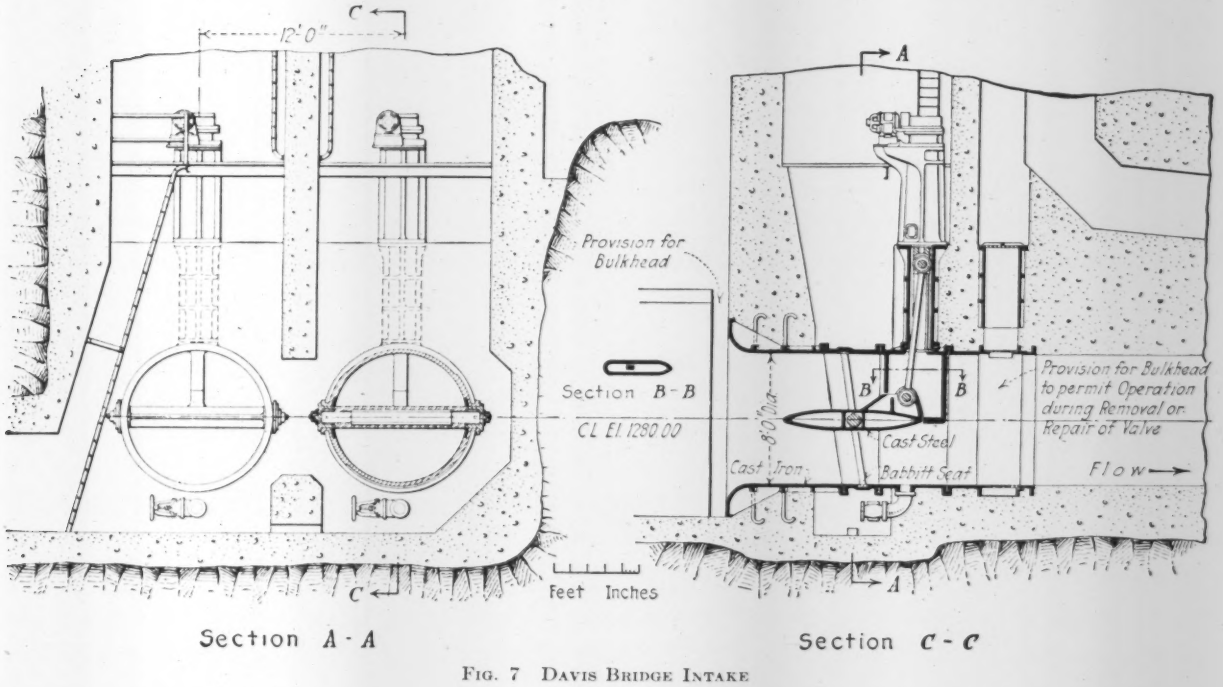


FIG. 7 DAVIS BRIDGE INTAKE

PENSTOCK AND TURBINE-INLET VALVES

When water is supplied to a hydro station through a single tunnel, conduit, or canal, which divides into several penstocks to feed individual units, it becomes necessary to provide means for complete shutdown from each unit separately to avoid shutting down whenever a single unit must be unwatered. This situation arises at plant No. 5, where the water is carried through three miles of conduits, to a simple surge tank, whence it passes into three steel penstocks; and at Davis Bridge, where the water is led through two and a half miles of 14-ft-diameter tunnel to a Johnson differential surge tank, immediately below which a plate-steel manifold distributes the water to three 9-ft-diameter steel penstocks about 600 ft. long.

The Davis Bridge situation brought up the question of whether to use valves at the top or bottom ends of the penstocks, or at both. Valves at the top would permit unwatering any penstock for inspection or maintenance work without interfering with the operation of other units, and also would facilitate a quick shutting off of flow in case of an accident to a penstock. On the other hand, valves at the top could not be used satisfactorily to reduce leakage during nightly shutdown periods, since this would involve too frequent draining of the penstocks which would probably involve increased corrosion. It would also be unsatisfactory as an operating proposition, for the time required to fill a penstock would render it impossible to put a unit on the line quickly in case an emergency arose. The decision finally arrived at was to use valves at both ends.

By actual measurement in the Davis Bridge plant the leakage through the wheel gates of each unit when first installed and in perfect adjustment was found to be about 9 sec-ft., an exceptionally low value which may be expected to increase considerably with time. The Davis Bridge units are operated in general on a 9-hour basis so that at least 5000 hr. per year, they are shut down. Assuming a leakage of 45 sec-ft. for the plant and an average head of 345 ft., the yearly loss of output would amount to something over 500,000 kw-hr. Since the Davis Bridge water is almost completely used, this is a real loss and the reduction of leakage by the use of valves at the turbine inlets is sufficient to justify the expenditure for such valves entirely aside from the time saved in getting into and out of a turbine when necessary.

The turbine-inlet valves used at the lower ends of the Davis

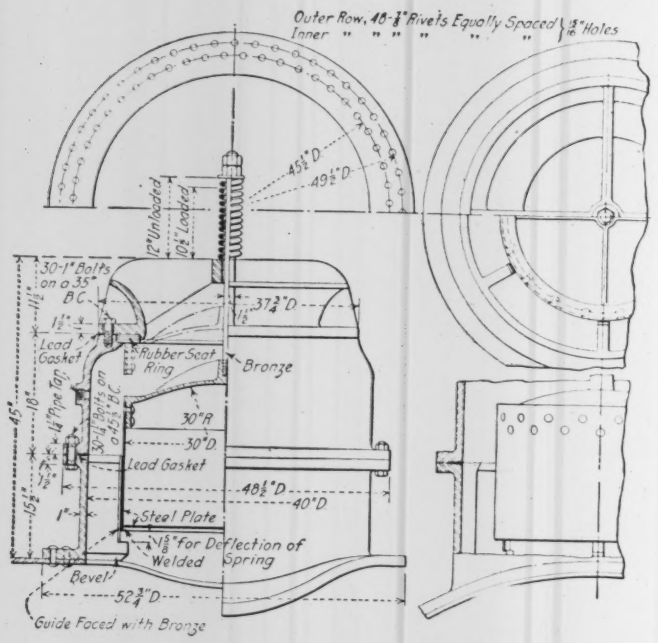


Fig. 8 Davis Bridge Air-Vent Valve

Bridge penstocks are of the ordinary butterfly type, provided with bronze seat and snap rings to increase tightness and controlled by hydraulic cylinders operating on penstock pressure. They have motor-operated pilot valves so that they can, if necessary, be closed or opened from the switchboard.

The valves at the top of the penstocks are of the same size and design as those at the intake, except that a single threaded stem is used instead of the stem and connecting rod adopted for the intake valves. The hoist is mounted on trunnions which permit it to follow the angular movement of the stem as the valve operates, tightness at the point where the stem passes through the body being obtained by means of a ball-and-socket joint provided with a cup leather. The valves are designed to close under free spouting velocity due to the full head of 160 ft. and are remote-controlled so that they can be closed from the power house in case of accident, the time required to close being about $3\frac{1}{2}$ min. It was not considered safe or desirable to fill the penstocks from a distance, since this might be done too rapidly if the valve opening were not carefully watched. To insure maximum dependability direct-current hoist motors were adopted, so wired that if necessary they can be operated from the station batteries. Ordinary operation is from the exciter system. The hoists are of rugged design and are provided with roller thrust bearings, which are preferable to ball bearings for heavy hoists. Like the valves at the intake the upper penstock valves are provided with babbitt seats and are extremely tight. The leakage at 150 ft. head does not exceed 2 gal. per min. each.

Air Valves and Vents

Whenever valves are installed at the upstream end of a pipe line it is necessary to guard carefully against the possibility of a dangerous vacuum or partial vacuum occurring if the line is drained with the valves closed. Provision for admitting air may be made either by means of open vents carried above the hydraulic gradient or by the use of air valves opening when the pressure drops below atmospheric. Both vents and air valves must be carefully protected from cold, and in most cases artificial heating is required to prevent freezing. Vents should be capable of admitting air at least fast enough to replace water drawn off through the turbine, and this without an undue drop in pressure to draw it in. If a turbine gate is suddenly opened with the penstock full and the headgate closed it may be impossible for the water column in a high vent pipe to accelerate quickly enough to prevent the formation of a serious vacuum, in which case the penstock would probably collapse before the vent emptied of water so that air could flow in.

With one exception, vents on the New England System are of the open type. Below the Davis Bridge penstock valves, however, such vents would have had to be nearly 200 ft. high to avoid overflow under load changes, and while these could have been run up and supported on the surge tank, the question of accelerating the water column and of preventing freezing resulted in the choice of air valves. For this service valves having a 24-in-diameter opening were developed and installed immediately below the penstock valves. They are shown in Fig. 8, and, as will be seen, are closed by a float

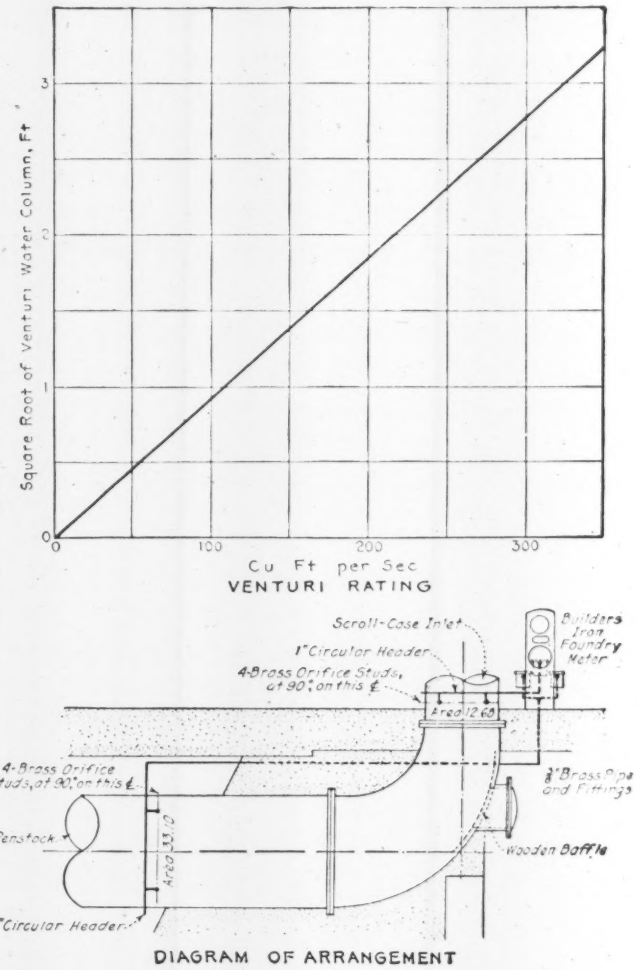


Fig. 9 Diagram of Plant No. 5 Venturi Installation

which rises as the penstocks fill. The bottom of the float is perforated to prevent collapse under the high pressure and also to reduce shock on closure by permitting the water to rise in the float and compress the entrapped air. The bronze float rides loosely in bronze-faced guides. When closed a flat machined seat on the float bears against a composition rubber and duck ring packing in the housing. The valve is tight and does not stick, though it opens with considerable suddenness when the pressure in the penstock drops. The opening is cushioned by the spring above the yoke at the top. Freezing is easily prevented by the use of a few electric space heaters.

DISCHARGE-MEASURING AND RECORDING DEVICES

Nothing is more important to economy of operation than trustworthy water records and reliable methods of checking turbine performance. In a high-head plant the use of venturi meters is usually possible, but the cost is considerable and the head loss by no means negligible. Several years ago the author's company investigated the feasibility of calibrating an inherited pressure drop at Plant No. 5, and the results were so satisfactory that the practice has been continued in later plants. The No. 5 penstocks are 6 ft. 6 in. in diameter at the lower end and the turbine inlets are 4 ft. 0 in. in diameter. The reduction in section takes place in a 90-deg. bend Fig. 9. Venturi connections were made at the inlet and outlet ends of this bend and by means of a mercury U-tube the pressure drop was determined at various discharges, the flow being measured by pitometer traverses. It was found that at full gate a pressure difference of slightly over 10 ft. of water was available and that this difference varied as a straight-line function of the square of the velocity. In other words, the pressure followed the regular venturi law and the constant was practically identical with that for a standard venturi. No difficulty was encountered in obtaining a venturi indicator-register-recorder instrument calibrated to suit the constant determined.

The average low-head plant does not afford any similar pressure drop which can be used for permanent water measurement. Any head loss, however, such as that through trash racks, intake gates, etc., can be calibrated for temporary use, and while this calibration will generally not hold for long it is likely to be very useful for the duration of a test, since it affords a means of taking a great number of observations in a limited time. Probably even more important is the possibility it offers of determining the characteristics of a turbine without any actual measurement of water whatever. On the assumption (apparently always correct) that the discharge is in proportion to the square root of the pressure drop it is possible to determine the shape of the efficiency curve of a wheel and plot, say, gate or power against percentage of maximum efficiency, and thus have a perfectly good guide as to the best range for operation, even though nothing whatever is known about actual efficiencies.

CONCLUSION

In conclusion the author wishes to state his belief that for best results as regards economy of operation in a hydro plant the following things are essential:

1 Proper design to reduce leakage and head losses to reasonable values. This does not mean, of course, that more money should be spent for this purpose than the savings will justify.

2 Suitable provisions for rapid and convenient inspection and repairs. This is even more important in the case of a hydro than of a steam station. A hydraulic turbine out of service with water spilling over the dam and a market for power means a total loss. If a steam turbine is out of commission it is at least possible to economize coal.

3 Continuous and reliable water records so that operating policies can be fixed with a clear comprehension of the discharge and storage that has been, is, and probably will be, available.

4 Convenient and trustworthy means of checking turbine performance, so that the condition of equipment can at all times be watched and a falling off in efficiency can at once be corrected.

Given these things, economical operation becomes merely a matter of exercising good judgment and the necessary vigilence to see that the desired policy is carried out.

The reliability of a hydro installation is largely determined before it ever goes into service. It demands suitable equipment, which has been selected with a proper regard to the service required, the use of rugged designs but with careful attention given to vital details, and the avoidance of complications. The author believes that strength and simplicity are the most important essentials of any kind of hydraulic equipment, and that these qualities are the keynotes of American design. Most of our troubles come when these ideals are either forgotten or disregarded.

Discussion

H. L. DOOLITTLE[1] corroborated the author's judgment in using valves at both upper and lower ends of the penstock. Remote control for opening penstocks should never be used, he said, due to the danger of loss of life in opening the upper gate while repair work was in progress. The increasing interest in the use of permanent water-measuring devices was gratifying as an indication of the realization of the importance of maintaining high operating efficiency.

F. R. Benson[2] complained of the lack of attention paid to the design of intake rack arrangements, saying that while thousands of dollars had been spent in attempting to raise the peak of hydraulic-turbine efficiency a fraction of one per cent, a partially clogged rack, cutting down the output several per cent, was considered a necessary evil. The value of the mechanical rake was no longer questioned, he wrote, and by means of it one rake might keep racks clear under conditions which would shut down a plant in spite of the efforts of thirty or forty men with hand rakes.

Robert W. Angus[3] commented on the discharge-measuring and recording devices mentioned by the author and computed the venturi coefficients. He thought some of the coefficients were too high and that they should not be constants, as indicated by the author, but should increase with velocity. The venturi-meter principle, he pointed out, could be used with a fair degree of accuracy in very abnormal cases.

Herman Will[4] called attention to the use of mechanical time recorders in connection with turbine governors by which the percentage of gate opening could be recorded.

W. M. White[5] told of measuring the quantity of water flowing to a turbine runner by means of the pressure drop in the pipe line; and expressed his satisfaction that the importance of accurate records of turbine performance had again been emphasized.

Arnold Pfau[6] said he wished to emphasize the author's statement that most troubles in hydroelectric plants came from disregarding strength and simplicity in design, and made a plea for higher standard in reviewing bids for hydraulic equipment.

O. H. Bundy[7] contributed a discussion dealing with his experiences with the six 72-in. disk-arm valves for discharge sluices which had been installed in the dam of the Sturgeon Pool plant of the United Hudson Electric Corporation, in which he showed how the leakage gradually diminished with use. He also offered some comments on erosion, in the concrete water passages, and on the design of racks, and emphatically approved of the author's recommendations that free-discharge valves be located at the downstream end of the waterway.

John F. Vaughan[8] cautioned against over confidence in automatic safety devices which relieved the operator of the necessity for eternal vigilance and responsibility.

At the request of Forrest Nagler,[9] the author described some experiences with valves discharging at high velocity into atmosphere. Turning to the measurement of capacity by the venturi principle, he confessed that he did not know why the coefficients as figured by Professor Angus should be so high unless perhaps the areas, which had been computed from drawings, were incorrect. However, the coefficient did not affect the use of the discharge records.

H. Birchard Taylor[10] said that the constant for measuring flow by the venturi principle made little difference so long as the actual calibration of the instrument was known.

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Principles of Metallurgy of Ferrous Metals for Mechanical Engineers[1]

VI—Alloy Steels

By LEON CAMMEN,[2] NEW YORK, N. Y.

ACCORDING to the definition adopted by the International Committee of the International Society for Testing Materials, alloy steels are "those which owe their properties chiefly to the presence of an element or elements other than carbon."

Thus steel containing, say, 0.30 per cent silicon is considered to be a carbon steel, while steel containing 3.20 silicon and developing certain special magnetic properties is called a silicon steel, is classed among alloy steels, and is extensively used in the construction of alternating-current transformers. There are many reasons why alloying elements are added to steels, and the action of some of them we understand fairly clearly, while the action of others is as yet quite obscure to us. We know, for example, why such elements as nickel are added to steel. The ultimate physical properties of carbon steels, as has been explained in Chapter IV, depend on the rate of cooling from the austenitic stage (from above the upper critical point Ar). If this cooling is extremely rapid as by quenching, the martensitic or troostitic stage is trapped and the metal becomes very hard. By cooling at slower rate the sorbitic stage is trapped, which gives the metal medium hardness and great toughness. Finally, by slow cooling the pearlitic stage is reached, and the metal is soft and highly ductile. It is obvious, however, that it is not always possible to achieve the first or second result. It is easy to cool quickly a thin piece of metal by quenching it in water, but if an attempt is made to cool in the same way a 22 × 22-in. ingot, the external layer will cool with extreme rapidity while the internal layer will stay hot for a very long time, and the result will be not a hardening of the ingot throughout but a hardening of the outside layer and gradual softening as we go inward, with a possibly soft heart. It is very likely also that the outside layer will be ruptured by the stresses induced by its effort to contract, while the inner part of the ingot retains its original dimensions. A judicious addition of nickel, however, produces structural changes in the steel giving a result equivalent to quenching of a thin piece of steel, and we know fairly well the mechanism of this process.

How other alloying materials act we can merely conjecture. For example, extremely small additions of vanadium of the order of 0.1 per cent produce material changes in the steel and substantially increase its ability to resist alternating stresses. It is extremely unlikely that the minute quantities of vanadium distributed throughout the mass of the steel are capable of producing such changes by entering into direct combination with either iron or carbon, and the general feeling is that vanadium acts as a catalyzer, producing changes by its mere presence that are entirely out of proportion to its mass. It is well, however, to remember that even if this surmise is correct, it does not really explain anything, because, even assuming that vanadium acts as a catalyzer, we have no information as to the reaction which it promotes.

It should be also clearly understood that some of the alloying elements are particularly valuable in that they either remove entirely or make harmless constituents which otherwise reduce the desirable properties of steel. Thus, the action of titanium is primarily to reduce the oxygen and nitrogen contents of steel and to impart to it thereby a closer grain structure. One of the reasons why zirconium is used in steel is to counteract the influence of the sulphur always present therein, and so on.

The steels with which the mechanical engineer has most to do are those containing as alloying elements nickel and chromium, singly or together, vanadium alone or in combination with the two others previously named, and the so-called "high-speed" tool steels characterized by substantial amounts of chromium and especially of tungsten. The mechanical engineer should, however, know something about the other alloy steels, as intensive research work is going on in many places and there is no doubt that many of the steels which have, relatively speaking, only a theoretical interest today, may become of practical importance in the near future.

Nickel Steel

This is one of the most important commercially of all the alloy steels. The introduction of nickel into carbon steel does not, according to some investigators, create a new constituent, although others have found that nickel and carbon form under certain conditions a carbide of the formula Ni_3C, which, however, breaks down very rapidly into nickel and free carbon. If nickel can replace some of the iron in iron carbide and then break down, this would account for the acceleration of graphitization due to the presence of nickel.

The most important effect of nickel in steel appears, however, to be in the direction of suppressing the transformation points. With low percentages of nickel all the critical points are lowered, and above 8 per cent nickel all the three points are united. We have seen in Chapter I that steel when passing through the Ar_2 critical point becomes magnetic and that this point for ordinary steel is located at a high temperature.

Nickel steels as well as nickel-iron alloys have a somewhat peculiar behavior in respect to the location of the critical points and appearance and disappearance of paramagnetism. If, for example, a 10 per cent nickel steel is heated, magnetism disappears at about 700 deg. cent. (1292 deg. fahr.). If now this steel is cooled magnetism does not reappear until about 400 deg. cent. (752 deg. fahr.). It has been noted before in this series of articles, namely, in Mechanical Engineering, August, 1925, p. 638, that the critical points Ar and Ac do not occur at exactly the same temperature, Ac being generally situated some 20 to 40 deg. higher than Ar. In the case of nickel steels, however, this difference between the Ac and Ar points may be as much as several hundred degrees. Up to a nickel content of 25 per cent the difference in temperature between the Ac and Ar points increases with the nickel content until at about 25 per cent of nickel the steel is non-magnetic even at 0 deg. cent. (32 deg. fahr.), but does become magnetic at lower temperatures. As the 25 per cent nickel content is exceeded the critical points at which magnetism disappears on heating and reappears on cooling gradually approach each other, until they come near merging. Steels in which there is the wide difference indicated above between the temperature at which magnetism disappears on heating and the temperature at which it reappears on cooling are called irreversible, while steels in which the two temperatures lie fairly closely together are called reversible.

There appears to be still some doubt as to whether nickel steel has superior resistance to repeated stresses. This is believed to be so by D. J. McAdam, Jr.[11]

Low-nickel or carbon pearlitic steels are formed with substantially the same structure as ordinary carbon steel, i.e., they consist of ferrite plus pearlite in the hypoeutectoid range, pearlite in the eutectoid range, and cementite plus pearlite above it, the only difference being that with the addition of nickel the eutectoid range is shifted to lower carbon-content percentages and is, for example, for 4 or 5 per cent nickel about 0.65 per cent carbon. When the nickel or carbon content is further increased, steels are obtained with the structure characterized upon ordinary unacceler-

[1] Sixth of a series of articles discussing the underlying physical and chemical processes involved in the metallurgy of iron and steel. The first, dealing with the physico-chemical properties of iron alloys, appeared in the May issue, p. 339, the second, on the crystalline structure of ferrous metals, in the June issue, p. 479, the third, on determination of the properties of metals, in the July issue, p. 559, the fourth on heat treatment, in the August issue, p. 637, and the fifth on cast irons in the September issue, p. 735.

[2] Associate Editor, Mechanical Engineering.

ated cooling by either martensite or troostite, depending upon the amount of nickel. With still greater amounts of nickel, so great as to suppress completely the transformation point, the austenitic structure is preserved down to room temperature. Such steels are sometimes called "polyhedral," because of the appearance of their structure under magnification. (Compare Fig. 122 on page 319, no. 2, Metallography, by Samuel L. Hoyt, McGraw-Hill Book Co., 1921.) Industrially the most important are the pearlitic steels; the martensitic steels are somewhat like quenched carbon steel, i.e., hard and brittle, and therefore of little use; the austentitic steels may become of value when more about them is known, but the great cost of their production is a serious obstacle. The average effect of nickel in amounts up to 8 per cent is given as follows by Abbott:[20]

0.01 per cent of Ni increases the elastic limit 40 lb. per sq. in.
0.01 per cent of Ni increases the tensile strength 42 lb. per sq. in.
0.01 per cent of Ni increases the reduction of area 0.005 per cent
0.01 per cent of Ni decreases the elongation 0.010 per cent.

From this it would appear that little is to be gained commercially by using nickel steel without previous heat treatment, and it is the latter that imparts to nickel steel its valuable properties.

The heat treatment to which nickel-carbon steel is submitted is usually limited to quenching from 850 deg. cent. (1562 deg. fahr.) and drawing at certain temperatures which vary.

The quenching should be from a temperature well above the critical range. Such a heat treatment increases the strength of the nickel steel enormously without materially lowering its ductility and toughness. It should be noted, however, that it is chiefly pearlitic steels with which the engineer has to do.

In these steels the percentage of nickel varies from 3 to 4 per cent. There is then another series of nickel steels with a very high percentage of nickel, of the order of 25 per cent, and carbon from 0.7 to 0.9 per cent. These steels have a great resistance to oxidation at high temperatures and are therefore used for valve stems for internal-combustion engines, low-temperature electrical-resistance wire, etc.

Two other alloys of nickel and iron are of interest, namely, invar, containing 36 per cent nickel and having a negligible coefficient of expansion at room temperature or slightly above it, which makes it valuable for the manufacture of measuring instruments; and platinite, having the same coefficient of expansion as glass and therefore used for lead-in wires in electric lamps.

Vanadium Steel

Because of its position in the periodic system of elements to the left of iron, vanadium has a great affinity for oxygen and carbon. It may therefore be used as a deoxidizer, but in addition it has in some way not yet clearly defined a degasifying effect on steel. Vanadium is capable of forming at least five carbides, depending on the temperature. The majority of these carbides are intensely hard. Because of this and also because of the very high price of vanadium no steels are made with more than 1 per cent of this metal, and even then the additions of vanadium are so proportioned that it is barely present in the final material, most of it disappearing by way of oxidation and otherwise in the manufacture of the steel. It does not appear that vanadium affects very materially the location of the critical points.

In structure, vanadium steel resembles carbon steel, except that when sufficient vanadium is present it produces what the French metallurgist Guillet called the "special constituent." This constituent appears in relief in the microsection beginning with additions of about 0.7 per cent of vanadium, and increases as the vanadium increases up to 3 per cent. Guillet believes that this special constituent is a carbide, as it is found only when carbon is present and increases in amount also as the carbon increases.

Vanadium in steel has one very important effect, namely, it apparently increases the ability of the steel to resist impact. It also raises noticeably the yield point and tensile strength, but its effect depends very materially on the heat treatment the steel receives.

Manganese Steel

Iron and manganese form a continuous series of solid solutions as shown by the constitutional diagram laid down by Rümelin and Fick,[16] the liquidus and solidus falling very close together. Contrary to what happens in the case of a simple binary eutectiferous system such as illustrated in MECHANICAL ENGINEERING, May, 1925, p. 341, the freezing of an iron-manganese mold proceeds in the following manner.[17] The velocity of crystallization of the molten metal is so great that solid solutions are simultaneously established with different contents of manganese. Because of this crystals appear in the metal obtained from the mold which are differently affected by etching reagents. It is also important to remember that manganese is chemically so close to iron that it may replace the latter without going into chemical combination with it. Manganese in manganese steel behaves exactly like the iron. It unites with it to form ferrite, and like iron takes up carbon to form carbides.

The effect of manganese on the critical points of steel was determined by Wüst, who found that every one per cent of manganese up to about 5 per cent lowers Ar_1 by 20 to 25 deg. cent. (68 to 79 deg. fahr.) and that the hysteresis is increased by a slightly greater amount. Above 5 per cent manganese no heat effect was noted on cooling. Rümelin and Fick, however, found the heat at 816 deg. cent. (1501 deg. fahr.) from 1 to 50 per cent manganese, and another over the same range falling from 767 deg. cent. (1412 deg. fahr.) to 711 deg. cent. (1312 deg. fahr.).

There are really two manganese steels. One is a pearlitic manganese steel. Since manganese forms a double carbide with cementite, the effect is to increase the tensile strength of the metal. Manganese steel containing 0.45 to 0.55 per cent carbon with about 1.25 per cent manganese has been successfully used in a heat-treated condition for rifle barrels. D. R. Bullens[18] claims that when properly made and suitably heat-treated, lower-carbon pearlitic manganese steels are not brittle, but the manganese appears to make the steel more sensitive to ill treatment. The other kind of manganese steel, much better known, is the so-called Hadfield high-manganese steel containing anywhere from 8 to 14 per cent manganese and specially heat-treated by quenching it in water from 900 to 1100 deg. cent. (1652 to 2012 deg. fahr.). This is so hard that it cannot be machined or drilled, and yet it may be indented by a hammer. It also possesses a peculiar flowage when used in crushing machinery. While it cannot be machined, it can be forged, but when this is done care must be taken not to overheat the metal. The first characteristics of manganese steel appear to be, however, its resistance to abrasion and its ability to deform plastically.

High-manganese steel has no critical points, so that in heat treatment the operator has to be guided by the relation of the carbide to the physical properties. Silica-manganese steel with 0.5 to 0.7 manganese and 1.25 to 1.50 silicon is used in tempered gears and springs. It has the disadvantage of being very sensitive to heat treatment. Upon suitable treatment it gives a tensile strength of from 195,000 to 230,000 lb. per sq. in. It is said to be used extensively in France and Germany, but less so in this country.

Chromium Steel

Chromium is used so extensively in the production of alloy steels and is so important in its action that it has been referred to as one of the key metals, i.e., metals vitally essential to the development of the industry. It may be used either alone, in which case it profoundly modifies the properties of ordinary carbon steel, or together with other alloying elements, such as nickel, tungsten, cobalt, vanadium, molybdenum, etc.

In the periodic system of elements chromium is located to the left of manganese and has therefore reactions with iron quite similar to those of manganese. It alloys with iron in all proportions and forms solid solutions. There are also indications that iron and chromium are capable of forming a chemical compound or compounds which profoundly affect the physical properties of the alloy. Chromium has also a very high affinity for carbon, forming with it several carbides remarkable for their hardness, ability to persist at very high temperature, and close crystallization. Chromium appears also to be capable of holding carbon in solution. As regards its influence on steel, it would appear that chromium in steel has the ability of opposing both the disintegration and the reconstruction of cementite, thereby producing changes in the critical points of steel, raising the Ac range and lowering the Ar range.

The microstructure of chromium was studied by Guillet, who worked out the diagram shown in Fig. 21. From this it would appear that chromium steels, depending upon the chromium and carbon content, may be pearlitic, martensitic, carbide-martensitic, and of the pure carbide type. It would appear, moreover, that the carbide present in the composition of these steels is a double carbide of iron and chromium, and that the carbon-martensitic steels are a transition product.

Because of the very high critical points of chromium steels they must be forged at a higher temperature than carbon steels of similar carbon content, and should it happen that the forging has been carried out at a temperature below the critical, chromium steels have to be subjected to a longer and more powerful anneal than carbon steels. There seem to be, however, reasons to believe that what Guillet refers to as martensitic steels are not true martensitic steels.

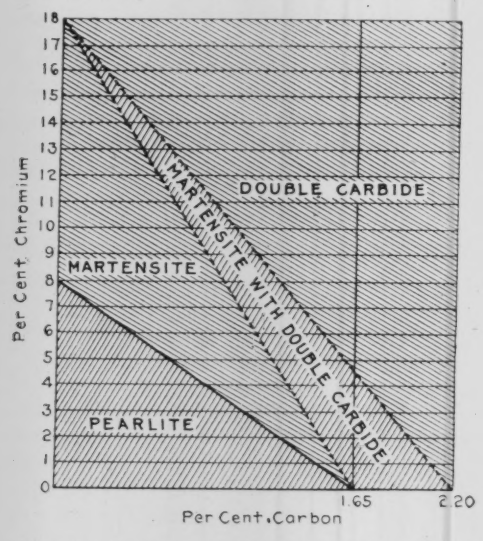

Fig. 21 Structural Diagram of Chromium Steel

Essentially, chromium steels behave in respect to slow anneal and rapid quench at high temperatures like the carbon steels. In annealed steels, carbides are predominant either in pearlitic form or as free carbides. The stable martensitic or polyhedral structures can be obtained only by very rapid cooling or quenching.

Unfortunately, comparatively little is known about the carbides present in chromium steel. Osmond claims that chromium in chromium steels may be present in one of the following three states: namely, as a solid solution of metallic chromium, as a combination with iron and carbon in the form of individual globules of carbide, or as the same carbide but in solid solution. Murakiam[1] believes that there are three ternary carbides of iron, chromium, and carbon, and that the hardness of chromium steels is due to the presence of a solid solution of the carbide Cr_3C_2 in a solution of iron and carbon.

It is important to remember that in addition to this, chromium and iron form a compound or combination which is generally referred to as "chromium ferrite" corresponding to the alpha iron in carbon steels.

According to Hoyt[2] the effect of the first additions of chromium is to make the steel finer grained and to disseminate the carbon. With further additions of chromium the steels resemble quenched carbon steels and contain martensite and troostite. Next comes the transition zone between the martensitic steels and the carbide steels. In this range chromium is added without materially affecting the microconstituents, the principal effect being a retardation of the crystallization of both ferrite and pearlite. More chromium produces a structure of hardened steels. On further increase in the chromium, the predominant martensite begins to disappear and polyhedra appear containing traces of fine-grained martensite in their interiors. Increasing the chromium still higher, up to 18 per cent and above, gives a polyhedral structure containing the carbide as small white crystals but exhibiting no martensitic structure. Steels with 0.8 per cent carbon are pearlitic up to about 3 per cent chromium. From 3 to 9 per cent chromium, a dark constituent (on etching) resembling troostite is present. A further increase of the chromium produces martensite as before, until with the final appearance of the martensite the steels show a polyhedral structure with white, structureless crystals, the carbide.

Thus far we have been dealing chiefly with the influence of chromium on the structure of the metal and its mechanical properties. Chromium has, however, another influence, and this is the increasing of the acid resistance of steels. This takes place in two ranges, within which the steel acquires what is known as the property of "passivity." According to Tammann,[3] ordinary iron becomes passive under certain conditions, as, for example, after immersion in nitric acid, through the fact that each iron atom on its surface is combined with an atom of oxygen and hence its connection with the iron atoms located below it has been interrupted. Mechanically, however, the film of oxidized iron does not adhere solidly but peels off and exposes to atmospheric and acid influences the rest of the metal. The chromium atoms stabilize this surface skin. As the content of chromium increases from 4 to 14 per cent the corrosion resistance of the carbon-free alloys with respect to dilute nitric acid increases very rapidly. From 14 to 20 per cent chromium this resistance increases rather slowly. This led to the development of the so-called "stainless" steels containing from 0.10 to 0.20 per cent carbon and from 10 to 14 per cent chromium, and stainless irons with approximately the same content of chromium but an even lower content of carbon. The value of these steels lies in the fact that when polished they are practically unaffected by organic acids. They are claimed also to have shown a high ability to resist erosion such as is encountered in blades of steam turbines. Another development along the same line is the production of heat-resisting chromium steels containing from 22 to 28 per cent chromium (Becket metal, Duraloy, Fahrite, etc.) and a variable content of carbon from 0.2 to 1 per cent (the lower percentages of carbon are used in pieces that have to be machined and the higher in those where excessive hardness is not objectionable). While the mechanical properties of these very high-carbon steels apparently are no better than those of corresponding straight carbon steels, they have an extremely high resistance to oxidation at high temperatures and may be repeatedly heated for very long periods at temperatures as high as 1800 deg. fahr., and in some cases 2000 deg. fahr.

Chromium steels of the following compositions are used.[4] Carbon up to about 0.35 per cent and chromium of about 0.5 per cent are employed where the steel has to be case-hardened. According to E. W. Ehn,[5] the chromium does not really affect the ability of steel to take the case but insures that the steel is free from oxides, and it is this freedom that is important in order to produce effective case-hardening. The same content of chromium with carbon between 0.35 and 0.50 gives after heat treatment about 15 per cent increase in tensile strength and 10 per cent increase in elastic limit with practically no loss in ductility as compared with straight carbon steels of the same carbon content. With the same content of chromium and high contents of carbon (up to about 1 per cent) the hardness increases still more. This kind of steel is extensively used for well bits and jars, drills, razors, etc. Chromium steels containing from 1.00 to 1.75 per cent chromium with carbon ranging from about 1 per cent up give some of the most important straight chromium steels. These are used in balls and races of ball bearings, crushing machinery, safes, tools, etc. The great advantage of this steel is that it can be made soft by annealing, say, 150 Brinell, and then can be hardened without distortion to 600 Brinell and over. The degree of hardness depends upon the distribution of the chromium-iron carbide, which is in turn controlled by the temperature-time elements in the heating and cooling phases. Chromium steels with about 2 per cent chromium are used in the manufacture of armor-piercing projectiles, crushers, cold rolls, drawing dies, etc.

Chrome-Nickel Steel

It is significant of the rapidity with which modern metallurgy has developed that chrome-nickel steels, which are such a vital element in the construction of modern machinery, especially automobiles, were first described by Guillet in 1905 in a paper before the Iron and Steel Institute. They were used for a considerable time previous to that, however.

Chromium and nickel when used jointly in steel and when properly treated appear to have the beneficial effects of both the chromium and the nickel without the disadvantages which are inherent in the use of either one separately. According to Dumas,[6] chromium produces a lowering of the critical points in irreversible

nickel steels (nickel content under 25 per cent) and a rise of the critical points in reversible nickel steels. According to Guillet,[7] five different structures are encountered in chrome-nickel-steels: namely, pearlitic with ferrite or carbide, purely martensitic, martensitic with inclusions of carbide, polyhedric, and polyhedric with inclusions of carbide. The transition from pearlitic to martensitic structures depends on the joint content of carbon, nickel, and chromium. An addition of chromium to martensitic nickel steel produces a formation of a certain amount of carbide and the appearance of polyhedra. In polyhedric nickel steels the addition of chromium produces a formation of carbide.

According to Hoyt,[8] chrome-nickel steels may be classified into three types which contain the following percentages of nickel and chromium: (1) nickel 0.5, chromium 0.7; (2) nickel 3.0, chromium 0.6 to 0.9; (3) nickel 3.5, chromium 1.5. Each type is used generally with three different carbon contents: (*a*) low carbon for case-hardening parts; (*b*) 0.35 carbon for shafting and similar parts; (*c*) 0.50 carbon for heat-treated gears, crankshafts, and transmissions of automobiles, motor trucks, etc. The critical points of these three grades of steels have been determined by the Bureau of Research of the Midvale Steel Company. The correct temperature for the final quench would be about 25 deg. cent. above the upper line.

In general the chrome-nickel steels are very similar to the nickel steels; but (*a*) they have higher elastic limits with about the same ductility; (*b*) they are harder than the nickel steels, especially when heat-treated; (*c*) they are more susceptible to heat treatment, and many of them harden considerably when cooled in the air; and (*d*) they possess dynamic toughness and resistance to fatiguing stresses in the highest degree, when properly treated; and on account of these properties they are not surpassed, as construction steels, by any other steels. It is interesting to note that the static properties of the low-nickel, low-chromium steels are nearly the same as those of high-nickel, high-chromium steels. However, the latter type is generally to be preferred for parts which are subjected to exceptionally severe service.

The low-chrome-nickel steels are less expensive than either the high-chrome-nickel steels or the $3^1/_2$-per cent nickel steel, and are therefore very generally used in sections up to one and one half inch round for parts which require high static properties but which do not require all that can be given in toughness and dynamic properties. They may be readily heat-treated to give the following properties: tensile strength 130,000 lb., elastic limit 105,000 lb., elongation 16 per cent, reduction of area 52 per cent, Brinell hardness 275, scleroscope hardness 47; or tensile strength 220,000 lb., elastic limit 185,000 lb., elongation 9 per cent, reduction of area 30 per cent, Brinell hardness 390, scleroscope hardness 62.

As regards the heat treatment of chrome-nickel steels and their mechanical properties, further information may be found in D. K. Bullens' book on Steel and Its Heat Treatment[9] and especially in an article by H. J. French.[10]

Molybdenum Steel

This is one of the comparatively new classes of steel and one on which more knowledge is desirable. It is not clear yet, for example, exactly how the introduction of molybdenum into steel affects the microstructure and location of the critical points. The general belief is, however, that the addition of small amounts, 0.20 to 0.75 per cent, of molybdenum to plain carbon, chromium, nickel, or chrome-nickel steel makes it possible to obtain by suitable heat treatment either higher elastic limit with the ductility, or higher ductility and especially a greater reduction of area at the same elastic limit; that such steels show greater depth of hardening (i.e., more uniform hardness throughout large sections); that they have a wide range of effective hardening temperatures, variation of which does not greatly alter the mechanical properties; and that to secure equivalent softening on tempering with other analogous steels, the draw temperature must be higher.[12] On the other hand, it is claimed that[13] one of the difficulties with molybdenum steel is its tendency toward dissociation, producing at times a badly balanced steel with foreign inclusions, difficulty in fabrication, and bad test and surface results.

In plain carbon steel molybdenum produces some improvement when the steel is heat-treated. In steel merely normalized molybdenum produces no improvement and may even make the steel worse. A comparison of chromium-molybdenum and chromium-vanadium steel, it is said, shows that there is no appreciable difference in physical properties between the two steels, and that 0.40 per cent molybdenum comes very close to giving practically the same results in a chromium steel as that obtained by the addition of 0.2 per cent vanadium. By increasing the molybdenum up to about 0.80 per cent the tensile strength may be raised without very great loss in ductility.

Hoyt presents the following figures which were obtained by the United Alloy Steel Corporation with a chrome-nickel-molybdenum steel such as was used for Liberty-engine crankshafts. The composition was: C 0.30; Mn 0.69; Cr 0.98; Ni 3.05; Mo 0.54. The bar was annealed at 1600 to 1605 deg. fahr., cooled in air, treated at 1475 deg. fahr. for half an hour and quenched in water, and then drawn at 1150 deg. fahr. for two hours and cooled in air. The figures given for this steel, taken from a publication of the Climax Molybdenum Company, are: elastic limit, 130,000 lb.; tensile strength, 142,000 lb.; elongation on two inches, 20.5 per cent; reduction of area, 65.0 per cent; Brinell number, 303; and Izod impact number, 67.

The advantages of these steels may be summed up somewhat as follows: (1) They can be produced with exceptionally high elastic limits without sacrificing ductility and toughness; (2) the quenching and tempering ranges are particularly large; (3) when annealed the steels are soft and easily machined; (4) when in the heat-treated condition the elastic limit can be left exceptionally high without sacrificing machinability. This latter point has particular significance in the commercial production of high-grade machine parts.

Gillett and Mack[14] also present data on a nickel-molybdenum steel containing 0.3 per cent carbon and 0.62 per cent molybdenum, and on several chrome-nickel-molybdenum and chrome-nickel-vanadium steels. A detailed study of the effect of molybdenum in nickel steels was recently made at the Woolwich Arsenal, England, on specimens cut from large forgings transverse to the direction of the forging. These tests, with some reservations, proved to be favorable to the molybdenum steel.

Molybdenum in high-speed steel did not prove successful and a cobalt-molybdenum high-speed steel containing no tungsten gave very poor results. On the other hand, molybdenum has been successfully used for automobile parts (Wills-St. Claire, and Studebaker cars), crankshafts of the Liberty motors made during the war by the Ford Motor Co., in blooming-mill rolls, forging dies, etc. Summing up the whole situation, it would appear that molybdenum does not accomplish anything that cannot be attained by the use of other metals, especially vanadium, but that there is a hope that molybdenum steel for the same effect will be cheaper than vanadium steel. Furthermore, some metallurgists are convinced that at the same cost as competing steels, there is a saving in the use of molybdenum steels due to the ease with which they are forged, heat-treated, and machined.

An automobile producer compared two lots of steering knuckles, totaling 20,000, made respectively from 0.30 per cent C, 1.25 per cent Ni, 0.60 per cent Cr; and from 0.30 per cent C, 0.20 per cent Mo, 0.60 per cent Cr. Both steels were heat-treated to 270–320 Brinell. The number of pieces machined per grind of tool, averaging all the operations of drilling, reaming, tapping, turning, etc., was over twice as many on the chromium-molybdenum as on the chromium-nickel. A similar comparison made on axle shafts showed a 36 per cent advantage for chromium-molybdenum. Still another comparison showed that chromium-molybdenum steel of 325 Brinell hardness machined as well as $3^1/_2$-per cent nickel steel of 270 Brinell hardness. On the two steels, heat-treated to the same tensile strength and Brinell hardness, the time of drilling, milling, and reaming was found to be about half as great with chromium-molybdenum as with $3^1/_2$-per cent nickel.

Other tests do not show so great an advantage for molybdenum steels, but the weight of evidence indicates that the machining properties of these steels constitute a real advantage.[15]

Uranium Steel

Several statements have been made to the effect that steel to which uranium has been added possesses particularly high resist-

ance to shock and is therefore of value for such purposes as armor plate. The great difficulty in producing uranium steel is the excessive loss of uranium in the making of the alloy, especially when attempts are made to add a high percentage of uranium, say, 5 per cent or more. There is also often marked segregation of uranium between the top and the bottom of the ingot. Uranium oxidizes very easily and therefore is easily lost. Keeney[21] states that steel makers using ferrouranium containing 15 to 30 per cent of uranium could not make the uranium stay in high-speed steel. J. A. Matthews[22] claims that while uranium has been proposed for use in high-speed tool steel, it is difficult to handle owing to the ease with which it is oxidized, and that steel to which it is added is more apt to show seams and defects than steel from which it is absent.

BORON STEEL

From all indications boron steel seems to be a very remarkable material of which little is known and on which less work has been done than it has deserved.

Several batches of boron steel were made up at the Bureau of Mines laboratory[23] and at the Bureau of Standards. The samples contained variously from 0.09 to 0.7 per cent boron. When the ingots were heated to the usual temperature for hot rolling the material was so brittle that it fell apart of its own weight. When the preheating temperature was materially lowered it became possible to roll the metal, but it still seemed to possess a considerable brittleness when hot (hot-shortness).

A very remarkable feature of the boron steels is that while the steel is freezing in the mold it has for a long period a peculiar consistency. As the molten metal starts to cool it is at first like molasses, then like dough or modeling wax. When the top of the ingot within the hot-top is poked to keep the hole open into the pipe for the escape of gas, the steel can be poked into any shape desired and the shape will be retained, just as in the case of pie crust when the border is fluted with the thumb.

This plasticity is retained far below the solidifying point of ordinary steel, apparently down to approximately the melting point of cast iron, or even below. It is probable that boron forms a very low-melting eutectic with carbon, so that the range between the liquidus and solidus of a steel containing boron is enormously increased, which would account for its plasticity. Boron also appears to confer marked hardness on steel. There is a good reason to believe that when more is known about this steel and its properties are better controlled, a method will be developed for pressing out of it such parts as gears, etc., especially where these parts are to be subjected only to moderate strains.

CERIUM STEEL

While in cast iron cerium increases the transverse strength and deflection, the metal freezing more slowly than untreated metal and giving less combined carbon, in steel it apparently increases the uniformity of structure. From tests made at the Bureau of Mines[3] it has been found that the ferrite network usually formed by manganese sulphide inclusions is absent in the cerium-treated samples. Bars taken from the lower part of the castings made from cerium-treated steels were cleaned and showed marked superiority in ductility. It is not known that any cerium steel is available in the market. Considerable work has been done, however, in utilizing cerium and cerium-lanthanum for the purpose of purifying steel by desulphurization and deoxidation. Cerium to the extent of 0.1 per cent has been used successfully for the deoxidation of monel metal. It does not appear that it is uniformly effective in steel, but it does appear that under certain conditions cerium acts as a desulphurizer. On the other hand, however, cerium steel has a dirtier appearance than steel without cerium and shows numberless tiny inclusions supposed to be cerium-oxide spots.

HIGH-SPEED TOOL STEEL

The progenitor of the modern high-speed tool steel was the Mushet air-hardening steel containing $1^1/_2$ per cent carbon, $^1/_2$ per cent manganese, and 7 to 8 per cent tungsten. In the beginning of the 90's chromium was substituted for manganese, which produced a steel containing 1 per cent carbon, 0.2 per cent manganese (Midvale Steel Co.), 2 per cent chromium, and 8 per cent tungsten. From 1894 to 1898 Taylor and White in connection with their investigation into the cutting of metals developed at the Bethlehem Steel Co. their famous high-speed tool steel which, at that time, contained $1^1/_2$ per cent carbon, 4 per cent chromium, and 8 per cent tungsten, and was subjected to a complicated heat treatment. It was later found that the heat treatment which they employed could be simplified a good deal and that better results were obtained by increasing the content of tungsten to between 13 and 18 per cent and by reducing the carbon to from 0.4 to 0.9 per cent. Later, cobalt was substituted in part for tungsten.

According to Guillet,[19] high-speed tool steel, depending upon the amounts of carbon, chromium, and tungsten, has the following microstructure: pearlite, martensite, martensite plus carbide, carbide in gamma iron, and carbide in troostite. In later investigations it was found that in order to obtain the best steel the sum of the carbon, chromium, and tungsten content must be high so that neither pearlite nor martensite forms on slow cooling, while the chromium must not be too high in order to prevent the formation of gamma iron. The end sought in high-speed steels of certain compositions on quenching is to avoid gamma iron and to produce fine-grained martensite containing no carbide.

The mechanism which would account for the cutting properties of high-speed steel and the part played in it by chromium is as yet quite uncertain. What is known, however, is that the hardness of a piece of high-speed tool steel as measured, for example, by the Brinell number, has very little to do with its ability as a cutting material. As early as 1905 Taylor pointed out the vast difference between hardness at room temperature and hardness and ability to retain hardness at a dull-red heat. The only method for judging high-speed steel is therefore the lathe test.

Vanadium was found to improve steel, apparently by cleansing it, but the addition of $^1/_2$ per cent of vanadium, from which great things were expected, did not prove greatly important. The introduction of cobalt (about 5 per cent) is said to give tool steel a more uniform performance, i.e., ability to work very well on mild steel, special steels, or cast iron. Cobalt steel is said to be particularly good for heavy roughing cuts. Additions of molybdenum do not produce a good material, while for uranium claims have been made which so far have not been entirely proven.

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Pumping-Engine Tests and Sewage Disposal at Milwaukee

Papers of Local and National Interest at Milwaukee Session—Description of Sewage-Disposal Plant and Study of Its Heat and Power Requirements—Duty Trials of a Pumping Engine

THE Milwaukee Committee provided a program of both local and National interest at the Spring Meeting of the A.S.M.E., in a Milwaukee Session, held on Tuesday evening, May 19, presided over by W. F. Durand, President of the Society. The speakers were all Milwaukee men. John Arthur Wilson, consulting chemist of the Sewage Commission of Milwaukee described the activated-sludge sewage-disposal plant of that city, and Robert Cramer, consulting engineer, and member of the Society, spoke on the heat and power requirements of such plants. The duty trials of the new vertical triple-expansion pumping engines at the Riverside Pumping Station were reported by Charles A. Cahill, member of the Society and consulting engineer, Cahill and Douglas. Fred H. Dorner, Chairman of the Executive Sub-Committee for the 1925 Spring Meeting, addressed the Society on the Economical Advantages of Cities Having Diversified Industries, giving a brief statistical review of Milwaukee's manufacturing activities and the advantages of their diversity. Two of the papers and the discussion on them follow.

The Activated-Sludge Sewage-Disposal Plant at Milwaukee

By JOHN ARTHUR WILSON,[1] MILWAUKEE, WIS.

THE Milwaukee Sewerage Commission is now completing the construction of a great sewage-disposal plant. When the Commission was organized in 1913, no method was known of separating the sanitary sewage and industrial wastes of a great city into (1) an effluent pure enough to be run safely into Lake Michigan, the source of its drinking water, and (2) dry solid matter in a form permitting its economical shipment and use as a fertilizer. By the end of 1919 it was demonstrated conclusively that the newly discovered activated-sludge process could be relied upon to give pure effluent, but the material removed from the sewage was a soupy sludge containing about 98 per cent of water, which resisted further dewatering. Calculations indicate that approximately 1,275,000 gal. of this sludge may be expected from the daily flow of 85,000,000 gal. of sewage from the 575,000 inhabitants of the district served. After years of intensive research, a method was finally devised for reducing this sludge to an attractive dry powder having a high fertilizer value.

The scope of this paper will be limited to a brief description of the general operation of the sewage-disposal plant and the fundamental principles involved.

Removal of Coarse Material

The sewage is carried to the disposal plant at Jones Island by a system of intercepting sewers terminating in four siphons under the harbor entrance, 75 ft. below water level, the two carrying high-level sewage having effective diameters of 72 in. and 48 in., and the two carrying low-level sewage, diameters of 54 in. and 42 in. The maximum capacity of the four siphons is 317,000,000 gal. daily.

All of the sewage passes first through coarse bar screens with bars spaced 2 in. in the low-level cage screens and $3^{11}/_{16}$ in. in the high-level rack screens. The coarsely screened sewage then flows through grit chambers where the velocity of flow is reduced to from 0.75 to 1.00 ft. per sec., permitting the heavier materials to settle out. The grit chambers consist of eight open concrete channels, each 8 ft. wide, 8 ft. deep, and 95 ft. long and capable of isolation from the rest for the purpose of draining and cleaning.

The sewage then passes through the fine screens, consisting of eight revolving drums, each 8 ft. in diameter by 8 ft. long and covered with manganese-bronze plates $^3/_{16}$ in. thick having 27.5 per cent of their surface slotted with openings $^3/_{32}$ in. by 2 in. Each screen is equipped with eight brushes for removing the screenings. The peripheral drum speed is from 8 to 12 ft. per min. and the brush travel 60 ft. per min.

Removal of Finely Divided Material

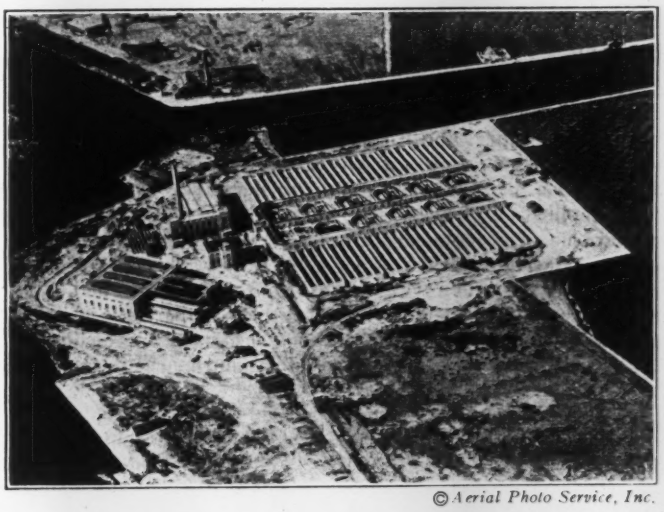
The removal of coarse materials involves only simple mechanical operations, but finely divided material that will not settle out upon standing and that cannot even be filtered, requires the operation known as the "activated-sludge process."

The screened sewage is passed first into a mixing channel 14 ft. wide and 11 ft. deep, through whose porous bottom air is forced to keep the liquor thoroughly agitated. At this point returned activated sludge is mixed with the sewage and the mixed liquor then flows on into the aeration tanks through feed channels, the flow being controlled by sluice gates.

The aeration tanks consist of twenty-four rectangular concrete tanks, each having two parallel compartments 236 ft. long and 22 ft. wide, with 15 ft. effective depth. In concrete containers resting on the bottom of each tank there are 2514 "filtros" plates, each 1 ft. square. Air compressed to 10 lb. per sq. in. is supplied to each container to be diffused through the filtros plates and allowed to bubble up through the mixed sewage and sludge as the mixture passes through the aeration tanks.

The flow is so regulated that it takes 6 hr. for the mixed liquor to travel through the aeration tanks, during which time 1.5 cu. ft. of air have been bubbled through it for each gallon of raw sewage entering the system. From the aeration tanks the mixed liquor flows into eleven large sedimentation tanks, each 98 ft. in diameter at the bottom, with an octagonal top and an effective depth of 15 ft. The mixed liquor enters the tank continuously and remains there for about 30 minutes, during which time the sludge settles out carrying with it practically all of the finely divided material. Pure water overflows into the effluent troughs at the top of each tank while the settled sludge is removed from the bottom by means of hydrostatic head.

At the top of each sedimentation tank there are three steel effluent troughs, each 32 in. wide,



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AIRPLANE VIEW OF THE MILWAUKEE SEWAGE-DISPOSAL PLANT

(In the right half of the picture are the aeration tanks separated into two groups with the sedimentation tanks between them. The building to the extreme left is the filter house and the drier house adjoins it to the right. Just above the filter house is the battery of grit chambers, to the right of which is the power house. The lone building between the power house and drier house is the laboratory building, which houses also the fine screens.)

[1] Consulting Chemist, Sewerage Commission of Milwaukee.

Contributed by the Milwaukee Spring Meeting Committee and presented at the Spring Meeting, Milwaukee, Wis., May 18 to 21, 1925, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. Abridged.

18 in. deep, and 98 ft. long. From these the effluent flows through two concrete conduits, each 8 ft. wide, 16 ft. deep, and 800 ft. long, to the lake outfall.

The sludge drawn off from the bottom of the sedimentation tanks, containing about 2 per cent of solid matter, flows through two parallel concrete conduits 6 ft. wide by 11 ft. 6 in. deep equipped with diffuser plates, and through a concrete tunnel 54 in. in diameter, to a sludge well. From here about 90 per cent of the sludge is pumped through a 48-in. cast-iron pipe, equipped with a venturi meter, to the channel where it is mixed with the incoming raw sewage. The sewage contains about 250 parts per million of suspended matter and enough returned sludge is added so as to increase this value ten times in the mixed liquor before it starts its journey through the aeration tanks. The 10 per cent of the sludge remaining in the sludge well represents the increase in volume of sludge derived from the sewage, which must be removed from the system and disposed of economically. It is pumped through an 18-in. cast-iron pipe to the filter house for dewatering.

The Mechanism of Aeration

When insoluble materials are suspended in a liquid, the rate of settling depends upon the size of the individual particles. The greater the size of a particle, the greater will be the rate of settling. The tiny particles present in sewage which will not settle at all can be made to settle rapidly by causing them to unite with each other to form relatively large particles.

The finely divided materials of sewage consist of animal or vegetable tissues presenting a relatively coarse structure. But through putrefaction and other chemical changes, these tissues have broken down into particles which will not settle out from the water upon standing. When oxygen is bubbled through neutral sewage, these tiny particles unite with each other, tending to form coarse particles, and if the bubbling is continued for several days, the particles become large enough to settle out fairly quickly upon standing.

If some of this settled sludge is now added to another portion of raw sewage and oxygen is bubbled through it, the tiny particles tend to combine with the larger ones rather than with each other. But the large particles are already of sufficient size to settle rapidly when the bubbling has ceased. Where two tiny particles combine, the double-size particle is still too small to settle out, but where one large and one tiny particle combine, the resulting particle will settle at least as quickly as the original large one.

This explains why it is desirable to mix sludge already capable of settling rapidly, and which is called "activated sludge," with the incoming raw sewage before aerating it. The time required for purification by settling is thus reduced from several days to six hours or less. In practice the activated sludge removes from 95 to 99 per cent of all the bacteria and suspended matter in the sewage, leaving a relatively pure effluent to be run into the lake.

The Sludge Problem

The successful disposal of the 1,275,000 gal. of waste activated sludge produced daily depended upon devising some means of reducing this to a dry material that could be shipped and sold as fertilizer. It was necessary first to reduce its water content from 98 to 80 per cent by some kind of filtering device. Only then would it be practical to dry it further by the application of heat. Unfortunately, no filter could be found capable of economically removing the bulk of the water from sludge produced in winter. This sludge was of such nature that it would quickly clog the filter cloth, rendering it impervious to water, even under pressure. The problem proved to be one largely of colloid chemistry.

Degree of Dispersion

The entire problem of the activated-sludge process appears to be one of regulating the degree of subdivision of the solid matter derived from the sewage and the degree of swelling, or water absorption, of the jelly-like constituents. Thus the further removal of water from the settled sludge depends upon the degree of dispersion of its solid matter. The particles formed in the aeration tanks may be large enough to settle sufficiently rapidly to permit continuous purification, but yet too fine to permit of dewatering by ordinary filter-pressing. In the summer, when the temperature of the sewage was above 20 deg. cent. (68 deg. fahr.), the sludge was always coarse enough to filter-press satisfactorily; but in the winter, when the temperature fell to as low as 7 deg. cent. (45 deg. fahr.), the finely divided sludge could not be filter-pressed without previous special chemical treatment.

Part of the sludge consists of organized jellies, such as fibrous protein matter. Success in filter-pressing depends not only upon the fineness of subdivision, but also upon the extent to which the jelly constituents are swollen with water. Fibrous protein matter may hold water in two very different ways: (1) In the interstices between the fibers, from which it may be forced out by application of relatively slight pressure, and (2) within the substance of the fibers themselves, from which the water can be removed mechanically only by the application of enormous pressure. The sludge will be in its best condition for filter-pressing when the individual particles are as coarse as possible and the water content of the jellies is a minimum. Happily, the conditions required for maximum tendency of particles of a protein dispersion to unite to form large aggregates are the same as those required for minimum swelling or water absorption of the protein jellies.

Relative Filtering Efficiency

Since the problem was the preparation of the sludge for filter-pressing, it seemed logical to use as a measurement of this condition the rate of filtration of the sludge under standard conditions. The apparatus used to make this measurement consists of a battery of uniform Buechner funnels set in filter flasks, all connected to one pump furnishing a high vacuum. As a standard, sludge was selected such that its rate of filter-pressing was considered on the border line between satisfactory and unsatisfactory. It was found that 500 cubic centimeters of such sludge diluted to contain 1 per cent of solid matter could be dewatered on a standard laboratory filter in just 20 min. This made it convenient to define the "relative filtering efficiency" of any sample of sludge as 2000 divided by the number of minutes required to filter 500 cc. of 1 per cent sludge through a standard filter. This gives the sludge which was only just satisfactory an arbitrary relative filtering efficiency of 100. Changing the size of the Buechner funnels or the kind of filter paper used merely changes the constant in the equation of proportionality between the laboratory and the large-scale plant operations. Several years of operation of large-scale filter presses show that the efficiency obtained for any given type of sludge is directly proportional to the value defined as relative filtering efficiency.

The next step was to study the effect of adding chemicals to the sludge and the effect of variable factors in the aeration tanks. Without special treatment it was found that the relative filtering efficiency of the sludge varied from a minimum of 5 in February to a maximum of 100 in September. Only in the late summer was it possible to filter-press the sludge satisfactorily without special chemical treatment. The problem was so to condition the February sludge as to make it filter at not less than 20 times its normal rate.

Effect of Adding Acid

The particles of activated sludge drawn from the settling tanks are negatively charged electrically. Much of the sludge consists of protein matter, which assumes a positive electrical charge in acid, and a negative charge in alkaline, solutions. With increasing charge, either positive or negative, the sludge tends to break up into smaller and smaller particles. Conversely, with decreasing charge, the particles tend to unite. The ideal situation is to make the electrical charge equal to zero. By adding increasing amounts of sulphuric acid to the sludge, the value of the negative electrical charge on the particles is decreased until it reaches zero. The exact point at which the sludge has no electrical charge is called its "isoelectric point."

The active constituent of all acid solutions is known as "hydrogen ion," and of all alkaline solutions as "hydroxide ion." Hydrogen ions and hydroxide ions unite to form water. In every aqueous solution the product of the concentrations of hydrogen ions and hydroxide ions has the same value. Knowing either, the other can be calculated, therefore it is convenient to express degrees of both acidity and alkalinity in terms of hydrogen-ion concentration. Because of the wide range, it is desirable to use the logarithm rather

than the actual value of the hydrogen-ion concentration. So the term "pH value" indicates "minus the logarithm of the number of grams of hydrogen ion per liter of solution." Thus a pH value of 3.00 indicates 0.001 gram H per liter. pH = 7 indicates a neutral solution, that is, one containing the same number of hydrogen ions as of hydroxide ions. Values above 7.00 represent increasing alkalinity, and values below 7.00 indicate increasing acidity.

The isoelectric point of activated sludge was found always to have the value pH = 3.4, regardless of apparent variations in composition of the raw sewage.

The effect of pH value on Milwaukee sludge has been measured daily for several years. In the following typical test, variable amounts of sulphuric acid or sodium hydroxide were added to samples of sludge when diluting to the standard content of solid matter, so as to give a wide range of pH values. Fig. 1 shows the results obtained in February, the upper curve representing a series filtered 15 min. after treating with acid or alkali and diluting and the lower curve an identical series filtered 24 hr. after treating. The sludge before treating had a pH value of 7.8; lower values indicate treatment with sulphuric acid and higher values treatment with sodium hydroxide.

The upper curve shows a clearly defined point of maximum at pH = 3.4. By adding enough sulphuric acid to lower the pH value from 7.8 to 3.4, we have brought about an increase in relative filtering efficiency from 5 to 25, a large amount, but still insufficient to permit satisfactory filtering.

The lower curve shows the marked changes undergone by the septic sludge upon standing for 24 hr. at different pH values. At all pH values the sludge showed a tendency to become redispersed, but this was most pronounced at a pH value of about 5.5. Further investigation indicated that septic action in activated sludge proceeds at the greatest rate at a pH value of 5.5. And that septic action undoes the work of aeration.

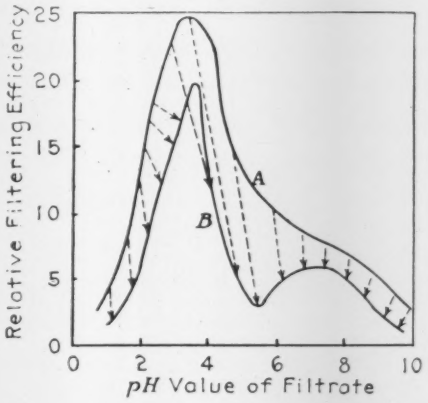

FIG. 1 EFFECT OF CHANGE OF PH VALUE UPON RELATIVE FILTERING EFFICIENCY OF WINTER SLUDGE

(Curve A represents series filtered 15 min. after treating with acid or alkali; curve B series filtered 24 hr. after treating. Dotted lines connect points representing samples receiving identical initial treatment.)

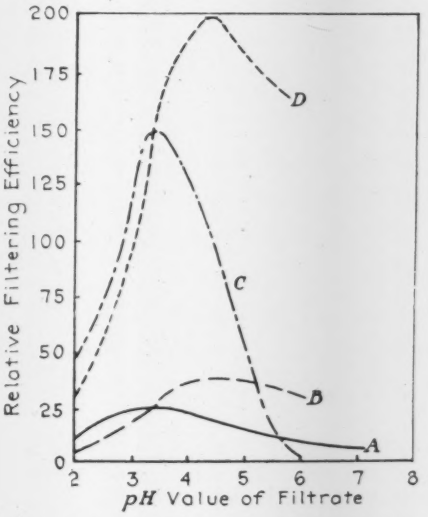

FIG. 2 EFFECT OF CHANGE OF PH VALUE UPON THE RELATIVE FILTERING EFFICIENCY OF SLUDGE TREATED IN DIFFERENT WAYS

(A, cold sludge treated with sulphuric acid; B, cold sludge treated with aluminum sulphate and sulphuric acid; C, sludge treated with sulphuric acid and heated to 88 deg. cent.; D, sludge treated with aluminum sulphate and sulphuric acid and heated to 88 deg. cent.)

EFFECT OF ADDING ALUMINUM SULPHATE

Sludge particles being negatively charged, increase in relative filtering efficiency was anticipated from the addition of the right amount of aluminum sulphate, which, when diluted, gives colloidal alumina particles with positive electrical charges.

The effect of adding aluminum sulphate to February sludge is shown in curve *B* of Fig. 2. For each gram of dry solid matter in the sludge, there was added 0.1 gram of aluminum sulphate, in diluting to standard condition. In this series only the pH value was varied, by the addition of sulphuric acid. It will be noted that a point of optimum is obtained at pH = 4.4. This may be explained by assuming that at this point the positive charge on the aluminum particles just balances the negative charge on the sludge. At pH values below 3.4, where both sludge and aluminum particles are positively charged, the aluminum apparently adds its charge to that of the sludge and the relative filtering efficiency is correspondingly decreased. The increase in efficiency obtained by the use of both aluminum sulphate and sulphuric acid is about 50 per cent greater than with sulphuric acid alone, but it is still insufficient to enable one to filter February sludge efficiently.

EFFECT OF TEMPERATURE

With colloidally dispersed material at its isoelectric point, coagulation is accelerated by heat. Fig. 3 shows the effect of increasing the temperature upon the relative filtering efficiency of February sludge with and without the addition of aluminum sulphate, but with the pH value adjusted to the optimum value in each case. Where sulphuric acid alone is used, an optimum value is obtained at a temperature of about 88 deg. cent. (190 deg. fahr.) above which a portion of sludge seems to disintegrate. With aluminum sulphate equal in quantity to one-tenth of the weight of dry solid matter in the sludge and sulphuric acid to make pH = 4.4, we get a curve increasing in value at all points up to the boiling point of water.

The effect of heating sludge at different pH values is shown in curves *C* and *D* of Fig. 2. Curve *C* shows the effect upon sludge treated with sulphuric acid only before heating, and curve *D* the effect upon sludge treated with aluminum sulphate and sulphuric acid before heating. In order to get a large increase in relative filtering efficiency by the application of heat, it is necessary to add the proper amount of acid before heating.

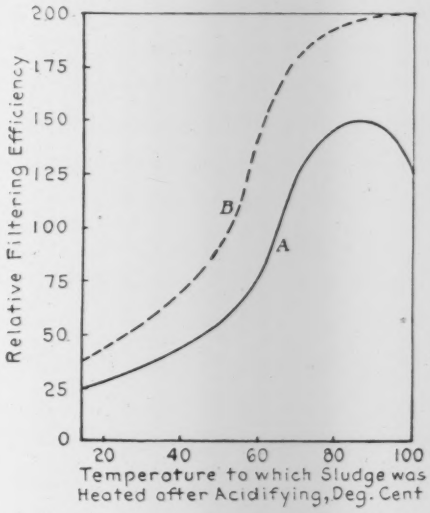

FIG. 3 EFFECT OF CHANGE OF TEMPERATURE UPON RELATIVE FILTERING EFFICIENCY OF WINTER SLUDGE

(A, treated with sulphuric acid to make pH = 3.4; B, treated with aluminum sulphate and sulphuric acid to make pH = 4.4.)

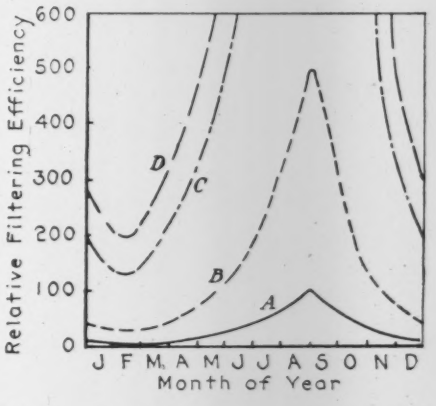

FIG. 4 SEASONAL CHANGES IN THE RELATIVE FILTERING EFFICIENCY OF ACTIVATED SLUDGE TREATED IN VARIOUS WAYS

(A, cold sludge, not treated; B, cold sludge treated with sulphuric acid to make pH = 3.4; C, sludge acidified to make pH = 3.4 and heated to 88 deg. cent.; D, sludge treated with aluminum sulphate and sulphuric acid to make pH = 4.4 and heated to 88 deg. cent.)

The application of both acid and heat increases the relative filtering efficiency of winter sludge from 5 to 150 and the further use of aluminum sulphate increases this value to 200. This brings winter sludge into a condition such that it filters twice as rapidly as untreated summer sludge. Thus one of the most difficult problems involved was solved by the application of colloid chemistry.

SEASONAL CHANGES

Fig. 4 shows the seasonal changes in the sludge treated in different ways. Each of the curves is subject to daily fluctuation, due to variations in temperature in the aeration tanks, septic action in the settling tanks, and other variable factors, but the general trend is unmistakable. The seasonal changes are actually due to the changes in temperature of the raw sewage. At the higher temperatures the building of sludge is accelerated, and the waste sludge shows a correspondingly greater relative filtering efficiency.

A very curious point to be noted is that any given treatment multiplies the relative filtering efficiency by a constant factor, regardless of initial condition.

APPLYING LABORATORY DATA TO PLANT PRACTICE

When the laboratory data were first applied to large-scale work, in which an Oliver continuous vacuum filter was used, there appeared to be a number of serious discrepancies. In one test, a given treatment of the sludge showed an increase of 100 per cent in relative efficiency in the laboratory tests, but with the same treatment of the sludge entering the filter tank, an increase of only 40 per cent was observed. It was noted, however, that the cake was thicker than was the case when the given treatment was omitted. This was clearly the reason for the discrepancy, for the relative filtering efficiency per unit quantity of sludge decreases when an increasing total amount of sludge is applied per unit of filter area. When the speed of the drum in the Oliver filter was increased to the point where it picked up the same quantity of sludge per square foot of filter cloth in the specially treated sludge as it had in the sludge not so treated, the increase in efficiency was then 100 per cent, exactly as in the laboratory tests. With increasing drum speed there is an increasing yield per unit of time but a thinner cake, which increases the difficulties at the take-off. The most efficient speed is that which gives a cake so thin that it can just be handled properly at the take off.

The blinding of the filter cloth proved troublesome. In the laboratory, fresh filter papers are used for each test, but on the big filter the same cloth is used over and over again. It frequently becomes "blind" and must be washed thoroughly before filtering can be continued. This variable degree of blindness has been the cause of discrepancies between laboratory and plant data. It was found, however, that the blinding of the filter cloth is a direct function of the quantity of sludge filtered since the last cleaning and is independent of the time required for the operation. In one test the total yield of sludge was plotted against the running time until blinding of the filter cloth occurred. The cloth was then cleaned and the test repeated except for tripling the drum speed. The quantity of sludge picked up per revolution of the drum was reduced to two-thirds, but the greater speed gave a net increase of about 100 per cent in yield per unit of time. But blinding occurred in both cases when a definite amount of sludge had been filtered. With drum speed and blinding under control, there is no difficulty in duplicating laboratory results in the plant.

DEWATERING THE SLUDGE

The waste activated sludge flows first into a 42,000-gal. acidification tank, equipped with diffuser plates and baffles, where it is treated with the amount of sulphuric acid necessary to give the filter effluent a pH value of 3.4.

The acidified sludge next passes into a heating system of ten Griscom-Russel heat exchangers having 19,500 sq. ft. of heating surface. It is then delivered to the series of twenty-four Oliver continuous vacuum filters, 11 ft. 6 in. in diameter and 14 ft. long, with 495 sq. ft. of effective filtering surface. The hot filter effluent is returned to the heat exchangers where it gives up its heat to the incoming sludge and then flows into the raw-sewage line where it enters the aeration tanks.

The sludge cake, while still hot from the filters, is conveyed by conveyors to six Atlas direct-indirect-heat continuous rotary driers, each 7 ft. in diameter by 60 ft. long, enclosed in brick settings. Because of the tendency for the sludge to ball up and dry irregularly in the driers, an equal quantity of previously dried sludge is mixed with the sludge cake from the filters in the screw conveyor. This lowers the moisture content sufficiently to prevent balling. As the drum revolves, the hot gases from the furnace, mixed with large volumes of outside air, circulate around and into the drum through numerous air valves attached to the drum shell. About 45 min. is allowed for the sludge to pass through the driers. The sludge cake coming from the filters has a water content of about 80 per cent. After it has passed through the driers its water content is only from 5 to 10 per cent.

PREPARATION AS FERTILIZER

The dried material is conveyed to rotary screens, from which the finer particles are transported to the storage house by belt conveyors. The coarser particles are passed through a pulverator and are then conveyed back and discharged into the wet-sludge-cake conveyor.

The material collected in the fine screenings of the sewage cannot be successfully handled with the sludge. It is estimated that there will be approximately 36 tons of screenings per day containing 85 per cent of water. These could be pressed and dried, but their value as fertilizer would then be practically nil. But, if they are digested before drying, their fertilizer value is about one-third greater than the sludge itself. It is therefore intended to put them through a horizontal digester 7 ft. in diameter by 40 ft. long, in which they are first subjected to 125 lb. steam pressure for 30 to 45 min. and then to a vacuum for 5 to 6 hr., which digests and dries them to 10 per cent moisture.

ANALYSIS OF SLUDGE

The following is a typical analysis of the dried sludge as a fertilizer:

	Per cent
Water	5.75
Total phosphoric acid	2.34
Total nitrogen, as ammonia	7.32
Water-insoluble nitrogen	6.65
Water-soluble nitrogen	0.67
Active water-insoluble organic nitrogen	3.94
Total available ammonia	4.61
Availability of nitrogen	62.97

When mixed with potash and phosphate the sludge becomes a complete and satisfactory fertilizer.

Discussion

F. W. MOHLMAN,[1] wrote that the Sanitary District of Chicago had had an Oliver filter in operation at the Calumet treatment works since June, 1923, and also operated other types of filters at the Des Plaines treatment works and at the corn products testing station, all filtering activated-sludge. Many laboratory tests as well as full-scale operating tests had been made of the pre-treatment of sludge with sulphuric acid, aluminium sulphate (filter alum), and heat.

Laboratory tests on Buechner funnels made at Calumet, Des Plaines, and Argo by the same chemist using identical methods, had shown decided superiority at all laboratories in the use of alum over acid with unheated sludge.

The results at the plants had shown some differences, it was true, but a general agreement as to the relative merits of acid and alum. Acid was of little or no value unless combined with excessive and expensive heating of the sludge. Acid cost more than alum in the amounts required by Chicago sludges. Dr. Wilson's results had indicated the reverse to be true with Milwaukee sludge. The interesting point of comparison was that each sludge, or each problem of sludge filtration, presented different conditions. It was his conviction that acid-heat treatment of Chicago sludge was neither advisable nor economical. He was endeavoring to find improved filter aids that would be superior to alum when used with cold or poorly activated sludge.

The filtration and drying of sludge was still a fertile field for investigation, and the future of the activated-sludge process depended to a considerable degree upon progress in reducing the cost of filtration and drying. The vacuum-type filter was undoubtedly the most satisfactory device that had yet been developed, but it must be admitted that the rates of filtration of activated

[1] Chief, Chicago Sanitary District, Chicago, Ill.

sludge were quite low when compared with the rates obtained with other types of sludges or suspensions. Pre-treatment of sludge in order to give greatly increased rates was the most likely field for improvement in reducing the cost of filtration. Dr. Wilson had done remarkable work in showing the application of principles of colloid chemistry to this difficult problem.

In reply to Dr. Mohlman the author said that he had come to the conclusion that the apparent discrepancies were due to insufficient data. He said that he had carried on tests extending over many years and had found that in order to plot quantitatively the efficiency of filtering against acidity it was necessary to have twenty points or the optimum point could not be determined. Inasmuch as Dr. Mohlman had used about six points, his optimum would be different. At Milwaukee, he said, it would be necessary to operate with very careful chemical control, and without a chemist in attendance constantly, operation would become inefficient.

The discussion of Mr. Cramer's address on the heat and power requirements of the sewage-disposal plant took the form of questions to which he made the following replies.

The burning of the sludge, although often discussed, had never been attempted because of its low heating value. Burning it to get rid of it had been considered but so far the necessity had not arisen.

The pounds of coal necessary to dry a ton of sludge could not be accurately stated. At Milwaukee the Atlas drier is used. In this dryer sludge is introduced into a drum which rotates slowly inside a furnace, and which is exposed partly to the heat of the furnace, and also to the heated products of combustion which pass through the sludge. Records kept in Chicago on this type of heater show that four pounds of water are evaporated per pound of coal fired, and it was upon this basis that the Milwaukee estimates were made. No doubt further study of this drying process would result in savings in fuel for drying. The temperature of the gases leaving these driers is 450 to 550 deg., occasionally reaching 750 to 900 deg.

Many methods of aeration other than forcing air through the sewage have been tried especially in England. In one of these systems the sewage is lifted by a paddle wheel and allowed to trickle back. Such plants have been successful in the south of England but have not been successful in Scotland on account of the accumulation of ice in winter. In another method the sewage is sprayed into the air by a centrifugal pump. The sludge formed this way, however, is not as good for filtering as that aerated by passing air through it.

Dr. Wilson also described an aerating system at Sheffield, England in which a wheel caused a wave motion which exposed a large surface to air contact. The sludge, however, could not be filtered by any mechanical process known. He said he knew of no system more economical than the one used at Milwaukee, when one considered that besides oxidizing the sludge it had also to be dewatered.

Duty Tests of Vertical Triple-Expansion Pumping Engines, Milwaukee, Wis.

By CHARLES A. CAHILL,[1] MILWAUKEE, WIS.

THE New Riverside Pumping Station was placed in operation with one unit on July 1, 1924, the second unit going into service in October. The two units in operation are known as "high-service" engines and operate against a head of 280 ft., while the third unit is for a head of 170 ft. and will pump into the Kilbourn Park Reservoir as soon as it is connected to the discharge line. This station was built to place six engines in service, three of which are now installed.

The city's growth is toward the high-service district and the low-service unit can be adapted for permanent operation on the high-service distribution by simply changing the pump plungers, neck rings, and stuffing boxes, the pump ends being constructed of ample strength for this service.

[1] Cons. Engr., Cahill & Douglass.

Contributed by the Milwaukee Spring Meeting Committee and presented at the Spring Meeting, Milwaukee, Wis., May 18 to 21, 1925, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

Milwaukee is the birthplace of the vertical triple-expansion pumping engine. The first engine was placed in service in the Tenth Street High-Service Station in 1886 and is ready to take its share of the load today, if required.

Milwaukee's first pumping engines in a municipally operated waterworks were placed in operation in 1874 and consisted of two 8,000,000-gal. engines of the Woolf compound duplex beam type, with bucket and plunger pumps. Steam was supplied at a working pressure of 50 lb. per sq. in. from Cornish boilers. A duty of 76,955,000 ft-lb. was obtained per 100 lb. of anthracite coal costing $7.60 per short ton. These engines were dismantled in 1913. They were built by the E. P. Allis Company, of Milwaukee. The late Thomas McMillen erected and started these engines, accepted the position as chief engineer of the Milwaukee Waterworks on October 1, 1873, and held his position continuously until his death

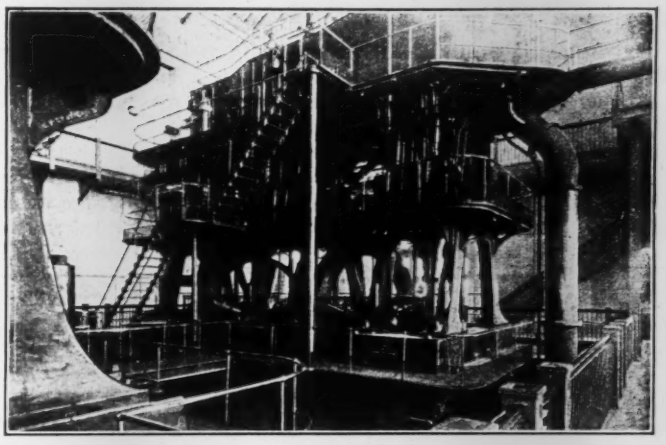

FIG. 1 RIVERSIDE PUMPING STATION, ENGINE A AS VIEWED FROM MAIN GALLERY

FIG. 2 RIVERSIDE PUMPING STATION, PUMPS OF ENGINE A

on May 1, 1920, a record almost without a parallel in municipal service.

In 1882 a 12,000,000-gal. pumping engine was furnished the city by the E. P. Allis Company. This engine was a vertical tandem compound beam type with bucket and plunger pump, the two steam cylinders and pump being vertically in line with the low-pressure cylinder above the high-pressure. Motion was transmitted to the flywheel through a beam. It is believed that this design was never repeated. It showed a duty of 104,802,431 ft-lb. per 100 lb. of coal.

In 1891 the city purchased from the E. P. Allis Company for the North Point Pumping Station an 18,000,000-gal. vertical triple-expansion pumping engine. This unit held the world's record for economy for some time. After 30 years of continuous service it has been thoroughly overhauled at a very nominal cost and will be placed in service for another 30 years' run, or until such advances are made in economical pumping machinery that it will not be profitable to operate it.

It is thus seen that since 1886 all pumping engines for waterworks service in Milwaukee have been of the vertical triple-expansion type and all with the exception of two have been built by the E. P. Allis Company or their successors, the Allis-Chalmers Manufacturing Company. They are a home product.

Steam pressures in Milwaukee pumping stations have gone

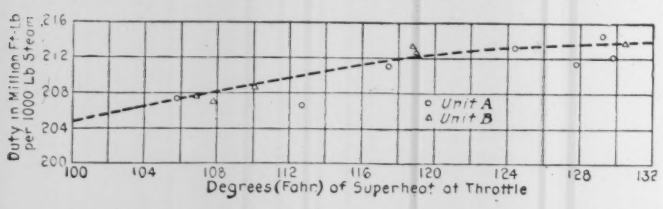

Fig. 3 Effect of Superheat on Economy of Vertical Triple-Expansion Engines, Units A and B, Riverside Pumping Station

from saturated steam at 50 lb. per sq. in. in 1874 to 210 lb. and 125 deg. superheat at the Riverside Station. In this respect our pumping stations have followed the universal rise in steam pressures and temperatures during the recent years, but have not as yet reached the so-called "super pressures."

Milwaukee has increased its pumping capacity from time to time with this type of pumping engine for the following simple reasons:

TABLE 1 PRINCIPAL DIMENSIONS OF THE PUMPING ENGINES INSTALLED IN 1924 AT THE RIVERSIDE PUMPING STATION, MILWAUKEE, WIS.
(Builder, Allis-Chalmers Mfg. Co., West Allis, Wis.)

		Engine A	Engine B	Engine C
1	Number of steam cylinders	3	3	3
2	Diameter of steam cylinders, in.	32-58-96	32-58-96	261-48-80
3	Diameter of piston rods, in.	6½-6½-8	6½-6½-8	6½-6½-8
4	Stroke of steam piston and water plungers, in.	60	60	60
5	Area of h. p. piston, head end, sq. in.	804.25	804.25	551.55
6	Area of i. p. piston, head end, sq. in.	2642.09	2642.09	1809.56
7	Area of l. p. piston, head end, sq. in.	7238.25	7238.25	5026.56
8	Net area h. p. piston, crank end, sq. in.	768.47	768.47	515.77
9	Net area i. p. piston, crank end, sq. in.	2606.31	2606.31	1773.78
10	Net area l. p. piston, crank end, sq. in.	7187.98	7187.98	4976.29
11	Proportion of gross area of pistons, head end	1:3.19:9	1:3.19:9	1:3.28:9.1
12	Proportion of net area of pistons, crank end	1:3.39:9.35	1:3.39:9.35	1:3.44:9.65
13	Number of water plungers (single-acting)	3	3	3
14	Diameter of each water plunger, in.	32⅞	32⅞	35
15	Area of each plunger, sq. in.	848.83	848.83	962.12
16	Displacement of 3 plungers per revolution, gal.	661.43	661.43	749.70
17	Number of valve chambers	6	6	6
18	Number of valves in each cage	35	35	35
19	Number of valve cages in each deck	7	7	7
20	Diameter of valve openings, in.	3⅜	3⅜	3⅜
21	Free valve area per valve, sq. in.	5.5	5.5	5.5
22	Net area of waterway through deck, sq. in.	1348	1348	1348
23	Ratio of net valve area to plunger area	1.59	1.59	1.40
24	Diameter of connected-air-pump bucket, in.	30	30	30
25	Stroke of connected air pump, in.	60	60	60
26	Type of condensers (surface)	Water-works	Water-works	Water-works
27	Number of condensers	1	1	1
28	Diameter of condenser tubes, in.	1¼	1¼	1¼
29	Inside diameter of condenser shell, in.	63	63	63
30	Cooling surface in condenser, sq. ft.	1500	1500	1200
31	Diameter of attached feed pump, in.	2¾	2¾	2¾
32	Diameter of attached air compressor, in.	2¾	2¾	2¾
33	Size of primary closed heater, sq. ft.	250	250	250
34	Shipping weight of unit complete, lb.	1,630,000	1,630,000	1,560,000
35	Weight of single heaviest piece, lb.	45,000	45,000	45,000
36	Revolutions per minute at rated capacity.	23.5	23.5	23.5
37	Rated capacity, million gal. per 24 hr.	22	22	25
38	Total head for which unit is designed, ft.	280	280	170
39	Plunger speed at rated capacity, ft. per min.	235	235	235
40	Diameter of main journals, in.	20	20	20
41	Length of main journals, in.	30	30	30
42	Diameter of h. p. and l. p. crankpins, in.	12	12	12
43	Length of h. p. and l. p. crankpins, in.	10	10	10
44	Diameter of i. p. crankpins, in.	18	18	18
45	Length of i. p. crankpins, in.	10	10	10
46	Diameter of crosshead pins, in.	11	11	11
47	Length of crosshead pins, in.	12	12	12
48	Diameter of plunger rods, in.	5⅛	5⅛	5⅛
49	Diameter of flywheel, ft.	18	18	18
50	Weight of each flywheel, lb.	65,000	65,000	55,000
51	Inside diameter of plunger barrel, in.	62	62	62
52	Thickness of metal in plunger barrels, in.	2½	2½	2½
53	Thickness of metal of discharge chambers, in.	2½	2½	2½
54	Inside diameter of suction chambers, in.	71½	71½	71½
55	Thickness of metal in suction chambers, in.	2½	2½	2½
56	Inside diameter of discharge chambers, in.	71½	71½	71½
57	Width of valve decks, in.	15	15	15
58	Diameter of suction nozzle, in.	42	42	42
59	Diameter of discharge nozzle, in.	42	42	42
60	Velocity through suction nozzle at rated capacity, ft. per sec.	3.53	3.53	4.02
61	Velocity through discharge nozzle at rated capacity, ft. per sec.	3.53	3.53	4.02
62	Size of secondary closed heater, sq. ft.	....	....	....
	2-pass Type D, Griscom-Russell No. 6 heater	Same	Same	Same

(*a*) *Extreme Reliability.* In the 39 years' operation of this type of pumping engine there has never been any anxiety on the part of the waterworks' officials or their staff as to its ability to do all it was guaranteed to do and much more. There has never been a breakdown.

(*b*) *High Economy.* The triple-expansion pumping engines at North Point have shown over a long period of years a remarkably high station duty over a wide range of capacity. They have also maintained this high duty over a long period of years.

(*c*) *Low Maintenance Cost.* The maintenance of this type of pumping equipment has been extraordinarily low in cost. Note the general repairs on the original 18,000,000-gal. unit at the North Point after a continuous run of 32 years.

The pumping engines at the Riverside Station are equipped with bleeder feedwater heaters taking steam from the low-pressure receiver. These heaters were not in use during the acceptance tests, the guarantees being made with exhaust heaters in service only. The bleeder feedwater heaters will increase the thermal efficiency of the units by from 6.5 to 7 per cent. The results of the acceptance tests on the Riverside units A and B are given in detail in Table 2, and the principal dimensions of these units in Table 1. The author believes from the acceptance tests on engines A and B at Riverside that this duty record has never been equaled by any steam-driven pumping engine to date.

The author believes that the continual increase in the thermal efficiency of this type of pumping engine is due to the very careful

TABLE 2 SUMMARY OF OFFICIAL TESTS OF PUMPS A AND B, RIVERSIDE PUMPING STATION, MILWAUKEE WATER WORKS[1]
(Object of Test: To determine fulfillment of duty guarantees.)

		Pump A	Pump B
	Contract Conditions		
1	Capacity, million gal. per 24 hr.	22	22
2	Total head in feet	280	280
3	Steam pressure at throttle, lb. per sq. in.	200	200
4	Speed in r.p.m.	23.5	23.5
5	Duty in million ft-lb. per 1000 lb. steam	200	200
6	Date of test (1925)	Jan. 13	Jan. 27
7	Duration in hours	10	10
	Average Steam Pressures		
8	Barometer in inches of mercury	29.61	30.07
9	Barometer in lb. per sq. in.	14.51	14.73
10	Steam pressure at throttle, lb. per sq. in. (gage)	199.02	200.50
11	Steam pressure at throttle, lb. per sq. in. (absolute)	213.53	215.23
12	Steam pressure in 1st receiver, lb. per sq. in. (gage)	35.70	35.93
13	Steam pressure in 2d receiver, lb. per sq. in. (gage)	—1.12	—1.06
14	Steam pressure in 2d receiver, lb. per sq. in. (absolute)	13.39	13.67
15	Vacuum in condenser in inches of mercury	28.58	28.93
16	Vacuum in condenser referred to 29.92-in. barometer	28.89	28.78
17	Absolute pressure in condenser, lb. per sq. in.	0.505	0.550
18	Absolute pressure corresponding to exhaust temperature, lb. per sq. in.	0.533	0.534
	Average Temperatures in Deg. Fahr.		
19	Steam at throttle	516.67	506.9
20	Saturated steam at test pressure	387.4	388.1
21	Superheat at throttle	129.2	118.8
22	First receiver	283.0	285.7
23	Condensate from air pump	38.2	38.0
24	Condensate after passing exhaust heater	76.0	77.6
25	Lake Michigan water at suction nozzle	36.2	36.0
26	Jacket and drain water (after entering tank)	189.7	189.0
27	Engine room, main floor	72.0	72.0
28	Exhaust steam (exhaust-pipe, top condenser)	81.7	81.73
	Water Pressures and Heads		
29	Average discharge pressure, lb. per sq. in.	117.13	115.73
30	Average suction lift, ft.	0.94	0.80
31	Differences in elevation, center to center of gages, ft.	3.61	3.66
32	Total head pumped against, ft.	121.68	120.19
	Total Quantities during Test		
33	Total condensate from air pump, lb.	90949	89759
34	Total jacket and receiver drains, lb.	10661	11381
35	Total steam used during test, lb.	101610	101140
36	Per cent of total steam used by jacket and drain	10.5	11.26
37	Total revolutions of engine	14071	14095
38	Water pumped per revolution, gal.	661.43	661.43
39	Total water pumped, million gal.	9.307	9.321
	Average Quantities during Test (Capacity)		
40	Average r.p.m.	23.45	23.49
41	Average piston speed, ft. per min.	234.5	234.9
42	Average rate of pumpage, million gal. per 24 hr.	22.335	22.365
43	Average rate of pumpage in per cent of normal	101.5	101.6
	Power		
44	Total work done during test, million ft-lb.	21,800	21,570
45	Average water horsepower from test data	1101.01	1089.38
46	Average indicated horsepower from cards	1121.88	1128.89
47	Average combined mechanical efficiency, per cent	98.14	96.5
48	Per cent of total i.hp. developed in h. p. cylinder	39.2	40.7
49	Per cent of total i.hp. developed in i. p. cylinder	28.7	28.0
50	Per cent of total i.hp. developed in l. p. cylinder	32.1	31.3

[1] All readings of gages, thermometers, etc., were taken at fifteen-minute intervals, during the tests, with great accuracy. All gages were calibrated before and after tests by means of dead-weight testers. Measurements of condensate and jacket drains was recorded on tested platform scales and carefully checked from hour to hour. Office calculations were carefully made and checked by the Allis-Chalmers Mfg. Co., engineers, working in conjunction with the city's engineers.

TABLE 2

		Pump A	Pump B
	Hourly Quantities (Average)		
51	Revolutions per hour	1407.1	1409.5
52	Total steam per hour, lb.	10161	10114
	Heat Data		
53	Total heat per lb. at throttle, B.t.u.	1273.8	1268.7
54	Gross heat supplied in test above 32 deg. fahr., million B.t.u.	129.431	128.316
55	Heat returned per lb. of jacket and drain water above 32 deg. fahr., B.t.u.	157.7	157.0
56	Total heat returned in jacket and drain water above 32 deg. fahr., million B.t.u.	1.681	1.787
57	Heat returned per lb. of condensate leaving exhaust heater, B.t.u.	44	45.6
58	Total heat returned in condensate, million B.t.u.	4.002	4.093
59	Total heat returned in jacket and drain water an condensate, million B.t.u.	5.683	5.880
60	Net total heat supplied during test, million B.t.u.	123.748	122.436
61	Net total heat supplied per lb. steam in test, B.t.u.	1217.9	1210.6
62	Heat available on Rankine cycle based on exhaust temperature, B.t.u.	398.3	396.3
63	Heat available on Rankine cycle under contract conditions based on exhaust temperature during test, B.t.u.	391.9	391.9
	Economy As Measured		
64	Duty per 1000 lb. of steam during test in million ft-lb.	214.55	213.27
65	Duty per million B.t.u. furnished, including heat returned in feedwater, million ft-lb.	176.16	176.17
66	Thermal efficiency of entire unit, per cent	22.65	22.65
67	Engine efficiency based on delivered work, per cent	69.1	69.2
68	Steam supplied per delivered hp-hr., lb.	9.23	9.29
69	Steam supplied per i.hp-hr., lb.	9.05	8.97
70	Net heat supplied per delivered hp-hr., B.t.u.	11,240	11,239
71	Net heat supplied per i.hp-hr., B.t.u.	11,030	10,846
	CONTRACT CONDITIONS		
	Economy Based on Correction Curve		
72	Duty per 1000 lb. of steam with steam at 200 lb. gage and 100 deg. fahr. superheat, million ft-lb.	204.6	204.6
73	Guaranteed duty per 1000 lb. of steam as per contract, million ft-lb.	200.0	200.0
	Economy Corrected as per Heat Available on Rankine Cycle		
74	Duty per 1000 lb. of steam corrected for heat available on Rankine cycle, million ft.lb.	211.0	210.5

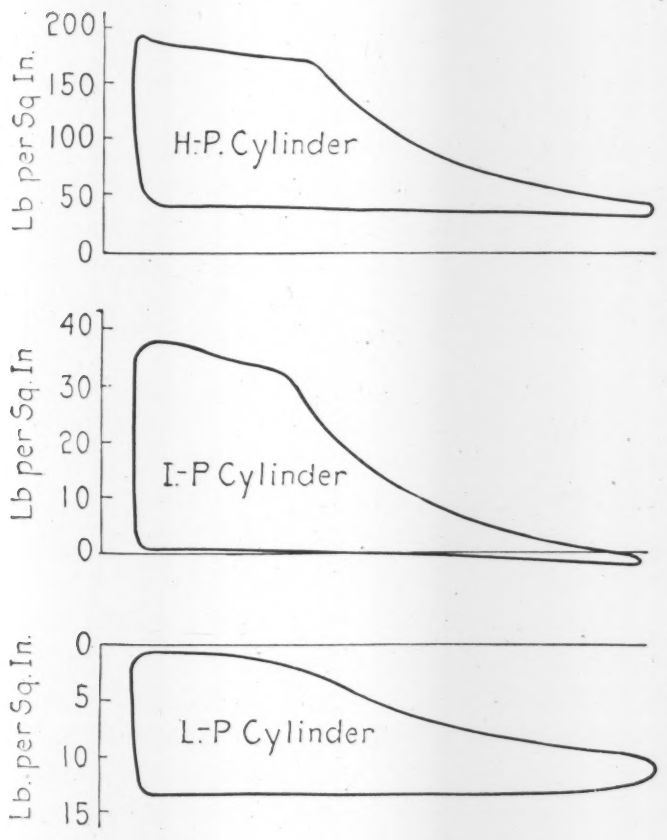

FIG. 4 TYPICAL HEAD-END INDICATOR CARDS, OFFICIAL TEST OF ENGINE A, RIVERSIDE PUMPING STATION

refinement of details and, of course, the use of higher pressures and temperatures, also from the fact that all engines of this type are invariably given a careful duty test.

It is interesting to note that the 18,000,000-gal. unit No. 4 at North Point Station set a new record in economy in 1892, and the results of recent tests at Riverside lead to the belief that after an interval of 33 years another record has been established with the same type of engine built by practically the same company.

Discussion

Geo. H. Barrus[1] wrote that he wished to concur with the statement that the superior economy realized was due "to the use of higher pressures and temperatures." Indeed, when compared with engines built by the same shops more than 30 years ago when saturated steam at 125 lb. pressure was used, it appeared to be due altogether to superheating and increased pressure, and not at all to "the very careful refinement of details" mentioned, or to any special merit of the engine itself.

From the chart of Fig. 4, the actual effect of superheating in this case could be computed; and he found that the 129 degrees of superheat in engine A added 51.6 million ft-lb. to the duty per 1000 lb. of steam. With saturated steam, therefore, the duty of unit A would have been 214.5 — 51.6 = 162.9 million ft-lb. per 1000 lb. of steam, or, proportionately, 133.6 million ft-lb. per million B.t.u.

From the indicator cards given, the effect of the increase of pressure on the duty above what it would have been at 125 lb. would be readily computed. He made the increase in power due to the increased pressure not less than 10 per cent. In other words, at 125 lb. pressure the duty would have been reduced one-eleventh, bringing it down to 133.6 — 12.1 = 121.5 ft-lb. per million B.t.u.

In 1895 he had conducted continuous duty trials on an engine of identically the same mechanical type and make as the one under consideration, having a capacity of 24 million gallons in 24 hours, which used saturated steam at 125 lb. pressure. The average duty for the whole run was 129.7 million ft-lb. per million B.t.u. It appeared from this comparison that so far as the merits of the engine itself were concerned, its thermal efficiency had decreased instead of increased as averred by the author.

He referred also to superior duties attained on other engines using saturated steam, so long ago as 1900. Two Boston engines each gave 156 million ft-lb. per million B.t.u., and one St. Louis engine reached 158 million ft-lb.

Robert W. Angus[2] wrote that the engines described in this paper were so nearly the same size as the one installed in Toronto in 1906 that the writer had compared the results which he had obtained in a test made March 26, 1906, with those given in the present paper.

The Toronto engine had steam cylinders 32, 60, and 90 in. diameter by 60 in. stroke, with single-acting water plungers each 32¼ in. diameter, while the Milwaukee engines had corresponding dimensions of 32, 58, and 96 in. by 60 in. with 32⅞ in. plungers.

The test speed of the Toronto engine was 19.83 r.p.m.; that of the Milwaukee engine A being 23.45; and the net water pressure in Toronto engine was 103.6 lb. per sq. in. as compared with 121.68 lb. per sq. in. for engine A. The Toronto machine was supplied with steam at 148.3 lb. with 1.22 per cent moisture. While the duty obtained in the Toronto engine was 172.3 millions of foot pounds per thousand pounds of steam used, the efficiency of the engine referred to the Rankine cycle was 69.2 per cent as compared with 69.1 per cent in engine A as given at item 67. The two machines had therefore given the same efficiency when compared on the same basis, and it would look as if this type of machine had reached its ultimate efficiency some years ago.

He called attention to the very high mechanical efficiency of 98.14 per cent on engine A and asked whether the indicator springs were carefully calibrated and what effort had been made to see that this figure represented correctly the true result.

While other waterworks plants were largely adopting the centrifugal pump, Milwaukee still adhered to the older type, and it was not clear why this had been done. It was true that higher duties had been obtained with vertical machines than with the centrifugal pumps, but the latter were being improved continually, and in a recent purchase by Toronto a duty of 157.5 million ft-lb. per thou-

[1] Expert and Consulting Steam Engineer, 12 Pemberton Sq., Boston, Mass. Mem. A.S.M.E.

[2] Professor of Mechanical Engineering, University of Toronto, Toronto, Can. Mem. A.S.M.E.

sand pounds of steam had been guaranteed with saturated steam at 150 lb. per sq. in., and the mechanical efficiency of centrifugal pumps of the capacity of the one described in this paper had reached 87 per cent.

The steam-turbine driven unit was better suited to the use of superheated steam than the vertical engine.

Against the lower duty must be put the lower first cost of the centrifugal unit, and he also believed that the maintenance cost of the latter unit was lower than that of the vertical machine. The pumps described had nearly 1500 valves, and the renewal of these was very expensive; and the other parts of the large machines required them to be kept in good condition to maintain their efficiency. In the centrifugal pump the only parts that ordinarily had to be renewed were the impeller and rings, and in over ten years' service these had shown no depreciation in Toronto. The cost of one of these impellers was approximately $1200 (not including customs duty), which would not go very far in the way of renewing the valves, etc., which would be necessary in the maintenance of the larger machine.

Loran D. Gayton[1] said that it was safe to state that this was probably a description of the last triple-expansion pumping engine that would be erected in this country. In Chicago there were many engines of this type. Two years ago, in designing a new plant, an exhaustive comparison had been made of all types, resulting in the choice of turbine-driven centrifugal pumps.

The Milwaukee engines had cost about $250,000 each. In Chicago an 80-million-gallon centrifugal pump for a head of 105 ft. and with a duty of 209 million ft-lb. had been bought for about $86,000. The building for triple-expansion units would cost about $294,000 more than one for turbine-driven centrifugals. There was no doubt about the efficiency of the triple-expansion engine but it was not great enough to offset the greater cost of buildings.

In answer to a question the author said that the steam-consumption figures in the table included the jacket steam. The Rankine-cycle efficiency was close to 70 per cent, and due to developments under way should be increased to 88 per cent. This he thought would tend not to eliminate the triple-expansion engine.

He said that a very careful study of types of pumping machinery had been made when considering the installation of the units. It had been found that if the units were run less than 3500 hours, the centrifugal pump would be the more economical. If they were run more than 3500 hours, the triple-expansion engine would be the more economical. This was without considering the building. As far as the building was concerned, however, he felt that little difference would be noticed as municipalities were not likely to save money on buildings.

Tests of a Uniflow Engine

Points Brought Out in the Discussion of a Spring Meeting Paper Presented by Geo. H. Barrus

IN A PAPER bearing the same title and presented at the A.S.M.E. Spring Meeting in Milwaukee, the author, Geo. H. Barrus,[2] gave a brief account of a series of tests which he had recently conducted on a four-cylinder vertical uniflow engine, non-condensing, driving a 400-kw. a.c. generator to which it was direct-connected, with loads ranging from one-quarter to full load, with steam pressures of 125 and 150 lb., atmospheric exhaust and 5 lb. back pressure, clearances of 17 and 22 per cent, and with saturated and superheated steam. The text of the paper appeared in the May issue of MECHANICAL ENGINEERING, p. 440.

In a written discussion which he contributed, James B. Stanwood[3] said that the non-condensing uniflow steam engine was fully established as a recognized element of direct-connected generating sets.

Mr. Barrus' tests were very instructive; they showed how the large amounts of steam shown by the indicator diagrams formed an important feature of the uniflow principle; this made it possible to employ early cut-offs and large ratios of expansion. In these tests the maximum cut-off was only 17 per cent; at these early cut-offs there were no large volumes of steam thrown away into the exhaust. The flat performance curve obtained was a result of these conditions, and was a very desirable feature not possessed by other types of engines. It was obvious that uniflow engines in order to secure capacity and economy for full load must use large cylinders, heavy frames, and generous running gears.

Although so great an improvement had been secured by uniflow cylinders, yet these tests showed a still more marked improvement when superheated steam was used. When saturated steam was employed at 125 lb. pressure under a wide range of load it had showed a nearly uniform performance of 20 lb. steam per i.hp-hr.; when the pressure was raised to 150 lb. there was an improvement in economy of about 10 per cent; and when to the 150 lb. pressure about 100 deg. fahr. of superheat was added, a further improvement of nearly 15 per cent was secured. The economical advantage of superheat was thus apparent.

A feature of this design of cylinder, as well as that of some other uniflow engines, consisted in passing the condensation of the jacket through the cylinder. What would have been the result if this condensate had been eliminated through a trap? The 66 per cent of the steam accounted for by the indicator diagram (shown by Table 6) undoubtedly included part if not all of this jacket condensate. When superheated steam was used, did it not prevent or materially check this jacket condensation?

The superheat as reported in these tests had been added by an independent superheater arranged to give 109 deg. fahr. at quarter load and 77 deg. fahr. at full load. In practice when the superheater was an integral part of a boiler the superheat usually increased as the load increased; this was the case with locomotives, and if it had been done in these tests the economy at higher loads would have been improved and the flat performance curve would have been more nearly horizontal.

Small installations of direct-connected generator sets had not often been used with superheaters. Mr. Stanwood had found that 72-in. by 18-ft. horizontal tubular boilers could be equipped with superheaters at a relatively small cost, and operated satisfactorily at little or no expense for upkeep. Two such boilers rated at 150 hp. were capable of developing 150 per cent of rating by using stokers or high stacks; they would evaporate 13,500 lb. of water into steam at 150 lb. working pressure with superheaters attached. Two of these boilers had ample capacity to operate the generating set upon which Mr. Barrus had conducted his experiments.

W. C. Brown[1] wrote that from 1876 to 1912, in which latter year Professor Stumpf first produced the uniflow engine, there had been no improvement in steam engines excepting in minor details, and notwithstanding the fact that the demand for steam engines had changed from those running at comparatively slow speed for driving mills, lineshafts, etc., to those operating at high speed for direct connection to electric generators, no successful increase in speed had been made in this country until the engine described in the paper had been brought out, although much more had been learned about inertia forces and the means for overcoming them and about materials capable of withstanding the higher speeds. Engine builders seemed to consider that it was the cost of the engine only that counted, and that, coupled with their objection to anything that required a change of patterns or standards, had retarded development and the reciprocating-engine business had suffered accordingly. Forcing electric generators to come to existing engine speeds was playing into the hands of the steam turbine.

In Mr. Brown's opinion, if the reciprocating engine was to hold its own it must be further developed along the lines indicated by the engine described in the paper.

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The first uniflow engine had been conceived from the fact that the turbine had shown the advantages of the steam passing in one direction so that the inlet end might always be maintained at a high temperature. Secondly, higher pressures and temperatures had been shown to be practical by the oil engine. Thirdly, multiple cylinders and higher speed had been copied from the gasoline engine. Fourthly, the next step to come was suggested by the steam turbine, namely, that of changing the pressure of the steam at the point of release into kinetic energy and utilizing the kinetic energy to draw steam from another cylinder so that the back pressure in the cylinder would be lower than the pressure at the point of exhaust.

The multi-cylinder engine lent itself to these last two advantages, because by multiplying the cylinders it was possible to increase the speed, thereby cutting in two the cost of the generator and greatly reducing the cost of the unit, and also making possible the utilization of the energy left in the steam at the point of release to reduce the back pressure in the cylinder so that without the use of a condenser a partial vacuum might be obtained, resulting in improved economy of operation.

One engine builder in the country had further utilized the multi-cylinder engine in a situation where part but not all of the exhaust steam was desired for heating purposes, one or more of the steam cylinders exhausting into the heating system and the others into the condenser.

Mr. Brown, after reading his prepared discussion, said, referring to a question brought up by Mr. Stanwood as to the use of steam traps for taking the condensate out of the jackets, that with superheated steam there was no condensate in the jackets where the steam passed rapidly over them, and with saturated steam there was so little that it seemed to do no harm to let it go into the cylinders.

The uniflow engine was different from the old type of jacketed engine in that the cylinder head, when once it was heated up, kept hot. There was no cold exhaust steam to cool it off, so that the temperature of the whole cylinder head was practically up to boiler temperature, and any loss in temperature in a cylinder was more than made up by the compression. In Germany Professor Nagel had shown that at the time the valve was opened the temperature of the steam in the cylinder was 200 to 300 deg. higher than that of the entering steam, so comparatively little cooling off took place, and as the steam passed over the surfaces very rapidly, there was practically no condensation in the jackets.

F. R. Low[1] said that it was very impressive that an engine of the kind described was able to replace in a single cylinder the elaborate paraphernalia that had been thought necessary some years ago with the multiple cylinders to get rid of the cylinder condensation. He thought it would be interesting to figure out what the effect of that simplification had been upon the overall efficiency.

James Tribe[2] said that he was interested in the uniflow principle, and had been convinced that there was a future for it, but he was concerned as to its efficiency relative to the Corliss engine when the required back pressure rose. That was to say, there was a tendency today to increase the back pressure so that the exhaust steam might be taken at higher pressures for process work. He had known a case recently where 100 lb. gage was required, and that meant that unless there was a corresponding increase in the initial pressure, the uniflow engine would probably not be any more efficient than the Corliss engine.

So far as he could see, the very fact of the high back pressure must necessarily delay the point of compression. Whereas in a uniflow engine the point of compression began at about one-tenth of the stroke, if the back pressure was very high it was easy to see that at the end of the compression the pressure will be above the initial boiler pressure, and so might do much damage.

It would be necessary with this high back pressure and delayed compression to install an auxiliary valve which would permit the compression point to be delayed to such an extent that the final pressure would not rise above the throttle pressure. But the introduction of such a special auxiliary valve would bring the uniflow back again practically to the original Corliss exhaust valve which was placed at the initial end of the cylinder, and thus put back again into the old position the condensation which came from the lower temperatures in the cylinder when fresh steam was admitted for its second stroke.

Charles A. Cahill[1] said that in his opinion the crowning excellence of the uniflow engine was the fact that it was the simplest engine built that could produce the economy it did produce. The next important thing was that it would run over an extremely wide range of load with almost a flat steam consumption for all loads.

In the matter of high back pressure it was of course necessary to use what was known as an "auxiliary," but only a little of it was brought back through the auxiliary exhaust valve to delay the pressure, so he did not think that disadvantage would work out as badly as Mr. Tribe feared it would.

Rudolph Wintzer[2] said that his company had built a large number of uniflow engines and a few years ago had installed a very large one in the Minneapolis district. In this case there was the question of getting power and heat for the mill. In the summer time the mill did not use any heat at all; in the winter time it needed heat and steam corresponding to about 1300 hp. The engine had auxiliary exhaust valves, and both sides would run either condensing or non-condensing. In the summer time both operated entirely condensing on the floor; when the heating season started the mill needed very little exhaust steam, and in that case the loads were shifted over to the condensing side, leaving very little to the side that was not condensing. In this way the mill always got exactly the amount of exhaust it required, and never wasted any exhaust steam through the roof.

An engine without an auxiliary exhaust valve, in order to get the proper compression, needed a certain speed for a certain back pressure and for a certain steam pressure. But as soon as this was changed later on, it would be necessary either to provide larger clearance spaces or lose some of the efficiency of the engine. With the auxiliary exhaust-valve arrangement, the engines could always be regulated. While the end of the exhaust was entirely governed by the auxiliary exhaust pipe, this compression could be changed while the engine was running.

Mr. Tribe, referring to Mr. Wintzer's closing remarks, said that whether there was an increased clearance space and increased volume and consequently an increased surface, or whether there was a larger auxiliary valve to delay the compression, in either case cooling surface was added to the initial end of the cylinder, resulting in a decrease in the steam or thermal efficiency.

In closing, the author wrote that in presenting the paper at the meeting he had called attention to one point regarding the 150-lb. series, saturated steam, Table 4, which he had noted after the paper had been printed, namely, that the percentages of "steam accounted for by diagram at cut off" on this series had averaged 12 per cent less than those shown on the 125-lb. series, Table 2, which were given in Table 6 of the paper. As a reason for this reduction, the author suspected that there had been some leakage of valves at the higher pressure which did not exist at the lower pressure of 125 lb. This might readily happen, for the valves were ground in for 125 lb. only, and not for 150 lb. If 10 per cent were taken as the loss in steam accounted for thus caused it seemed probable that, with equal conditions as to leakage, the steam consumption in the 150-lb. series, saturated steam, would have been reduced to the following quantities:

Load, kw.	100	200	300	400
Lb. per i.hp-hr.	16.5	16.5	16.0	17.6

and in the 150-lb. series, superheated steam, to those given below:

Load, kw.	100	200	300	400
Lb. per i.hp-hr.	13.5	14.1	15.1	15.8

It might be well also to note that a comparison of the results of the four tests given in Table 4 revealed a slight inconsistency in the test at 300 kw. load. The indicated horsepower given was obviously somewhat erroneous, being too large, and this made the water per i.hp-hr. too small. A consumption of 19.2 lb. per i.hp-hr. instead of 18.1, was a more consistent figure, and this would likewise change the figure 16 lb. above given to 16.9 lb.

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SURVEY OF ENGINEERING PROGRESS

A Review of Attainment in Mechanical Engineering and Related Fields

New Data and Tendencies in the Technology of Firing

A VERY lively development is going on in the technology of firing, caused partly by the efforts to burn what has in the past been considered low-grade fuel and to increase the efficiency and economy of the firing plant. The efforts in this connection are first of all directed toward the investigation of processes of combustion with a view to utilizing the knowledge thus obtained for the design of better furnaces. Before and during the war important work in the burning of low-grade fuel was done in the direction of the application of under draft, auxiliary air admission, and air preheating. When, after the war, contact was resumed with the United States of America, efforts were made in Germany, as across the ocean, to employ a larger combustion chamber and to give greater freedom to the furnace builder as regards the materials of construction.

All of these developments are only the beginning. What is lacking is a clearer conception of the necessary experimental work.

The present paper is limited to a general consideration of this problem with respect to solid fuels only and not including pulverized coal.

Composition of Solid Fuel and Its Influence on Combustion

Several publications have recently pointed out the great importance of free (i.e., not combined with oxygen) hydrogen and its relation to the fixed and volatile carbon in the fuel. In order to understand the following a brief presentation of our modern knowledge on the subject may be necessary.

Depending on their age, coals have different compositions as regards carbon, hydrogen, and oxygen. This has been treated in detail in a book by Brookmann published by the Mine Owners' Association in Essen (Verein für die bergbaulichen Interessen). Fig. 1 has been plotted from the data contained in Brookmann's work; in this the abscissas represent the content of volatile constituents, while the ordinates show the percentage content of the individual constituents in the fuel (free from water and ash). From the graphs in the figure it appears that the lines of hydrogen content and of the fixed carbon not combined with hydrogen have fairly regular shapes. The hydrogen content in particular in all solid fuels runs about 5 per cent, and falls below that value only in lean fuels. The fixed carbon increases regularly with the age of the coal. The available hydrogen (H−O/8) is highest in coals with average gas content, namely, the rich coals, and falls off in gas coals to a very flat line and in lean coals to a somewhat sharper line.

A peculiar behavior is shown by lines for carbon and oxygen, but all of these lines agree to the extent that as the oxygen content falls off rapidly, the carbon content increases strongly. One cannot help recognizing, however, that as the age of the coal increases, the oxygen content falls off. A comparison of lines for total carbon and for fixed carbon shows that irregularity in carbon content is due exclusively to the carbon contained in the volatile constituents. The line (H−O/8)/C, i.e., the ratio of the free hydrogen to carbon, is of practical importance, because this ratio affects combustion tremendously. This ratio is most favorable for coals of average gas content, namely, rich coals; it falls off slightly in the case of gas coals and falls off strongly in coals poor in gas. Because of this, as has been found, in practice, the combustion of gas coals is easier and better than that of lean coals. The most favorable ratio prevails in the case of rich coals, which, as a rule, give the best results in practical work. The ratio $\frac{H - O/8}{C_{vol.}}$ is of importance from the point of view of the firing process of the volatile constituents. In passing from lean coals to gas coals this ratio increases at first very strongly and then less so. This course of the graphs would appear to indicate that the gas developed from the fuel burns faster with lean coals than with gas coals.

Fig. 2 is also quite instructive in that it shows the amounts of the various constituents present in the different kinds of coal. While in lignite fixed carbon, combustible gases, and incombustible gases are present in about equal quantities, coke and anthracite consist practically exclusively of carbon.

Recently there has been a growing conviction that the oxygen present in fuels is not necessarily combined with hydrogen. It may be present in coals in solid form. Once combustion is started it makes its appearance, a fact which would explain the easy inflammability of young coals having a high content of oxygen.

The heat value is materially favored by the presence of hydrogen, while presence of oxygen reduces it very much. The highest heating value belongs therefore not to rich but to lean coals. It is well known that the temperature of combustion depends on the heating value and the air requirements, but since this is itself in a way dependent on the heating value, the combined variations in air supply and heating value may result in equally high combustion temperatures, at least for the medium- and high-grade fuels. Where the water content is high, there is a reduction of the temperature of combustion due to the evaporation of the water and the superheating of the steam.

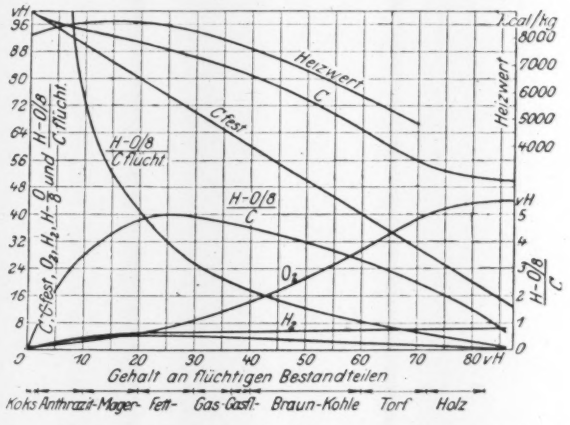

FIG. 1 COMPOSITION OF COALS (FREE FROM WATER AND ASH)
(*vH* = per cent, *heizwert* = heating value, *C* = carbon, *C fest* = fixed carbon, *C flücht* = volatile carbon, *Gehalt an flüchtigen Bestandteilen* = content in volatile constituents, *Koks* = coke, *Anthrazit* = anthracite, *Mager* = lean, *Fett* = rich, *Gas-Gasfl.* = gas coal *Braunkohle* = lignite, *Torf* = peat, *Holz* = wood.)

No attempt will be made here to elucidate whether the carbon burns to CO_2 or to CO. Recent investigations would appear to indicate that at first both take place and that the amounts burning to CO_2 and CO depend on the temperature and water-vapor content. The initial formation of carbon monoxide is, however, under conditions existing in practice, so slight that it may be neglected; and in general one may assume that the solid carbon is burned on the grate and the volatile constituents above the grate.

Efficiency of Firing

Hitherto entirely too little attention has been paid to firing processes, and yet it is necessary to increase to the utmost the efficiency of firing. Lately leading engineers have emphasized more and more that apart from the total efficiency of the plant the efficiency of firing as such must be increased. Ebel's tests with low-grade fuels have shown within what an extraordinarily wide range the efficiency of firing may vary, he having found efficiencies ranging from 78 to 96 per cent.

In order to judge of a firing plant, it is necessary to go further than data covering the size of the firing chamber and grate area, and separate the combustion processes of the volatile constituents from those of the coke residue, since the volatile constituents burn above the grate, i.e., in the combustion chamber proper, and they burn in a floating state, while the coke residue burns on the grate in a stationary state. The combustion of the volatile constituents is closely limited as to time by the interval they remain floating previous to contact with the heating surface, and lasts as a rule only a few seconds As far as the coke residue is concerned, the time element may vary widely, depending on the velocity of the draft and the length and rate of travel of the grate, and may last from $^1/_2$ to $1^1/_2$ hr. It is an advantage for the coke residue to have air rich in oxygen, but it is a disadvantage not to have the air pre-

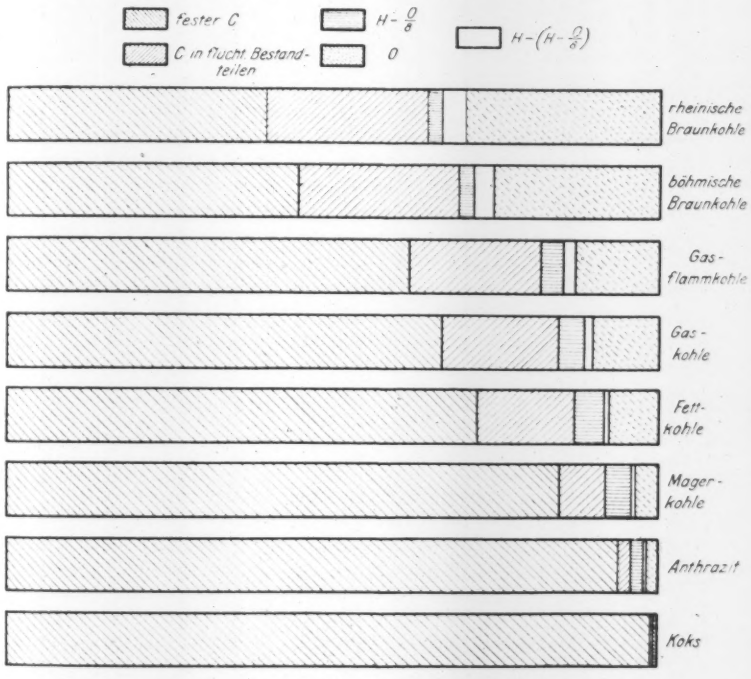

FIG. 2 COMPOSITION OF VARIOUS TYPES OF COAL

(*Fester C* = fixed carbon, *C in flücht. Bestandteilen* = carbon in volatile constituents, *rheinische Braunkohle* = Rhenian lignite, *Böhmische Braunkohle* = Czechian lignite, *Gas-flammkohle* = gas flaming coals, *Gaskohle* = gas coals, *Fettkohle* = rich coals, *Magerkohle* = lean coals, *Anthrazit* = anthracite, *Koks* = coke.)

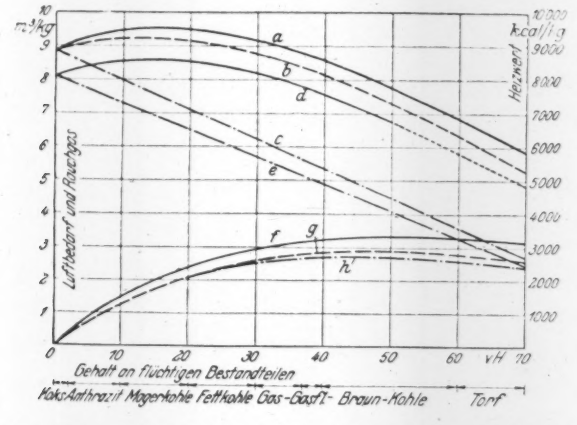

FIG. 3 AIR REQUIREMENTS, FLUE-GAS AMOUNTS, AND HEATING VALUES OF VARIOUS PURE COALS (ASH- AND WATER-FREE)

(*Luftbedarf und Rauchgas* = air required and flue gas, *Heizwert* = heating value, *vH* = per cent, *Gehalt an flüchtigen Bestandteilen* = content of volatile constituents, *Koks* = coke, *Anthrazit* = anthracite, *Magerkohle* = lean coals, *Fettkohle* = rich coals, *Gas-Gasfl.* = gas coal, *Braunkohle* = lignite, *Torf* = peat.)

heated. The volatile constituents get air that is preheated, but which is more or less deprived of its oxygen.

AIR REQUIREMENTS OF THE FUELS, THEIR HEATING VALUE, AND AMOUNT OF FLUE GASES

To what an extraordinary extent the air requirements of fuels may vary is shown in Fig. 3. The total air requirements for pure coals naturally fall off with the oxygen content of the coal and are generally considerably lower with younger coals than with the older varieties. They are highest for lean coals and decrease again somewhat for anthracites and coke. The air requirements for the combustion of coke residues increase uniformly with the age of the coal, while the air requirements for the combustion of

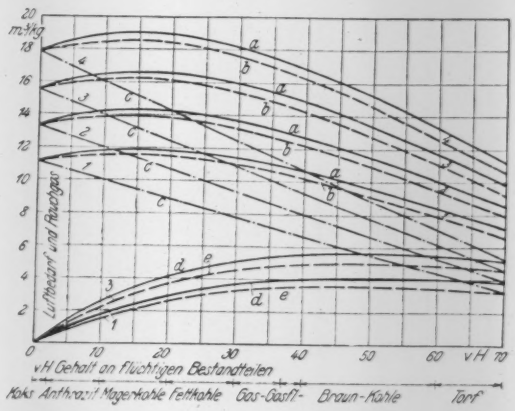
m³/kg
Luftbedarf und Rauchgas
vH Gehalt an flüchtigen Bestandteilen
Koks Anthrazit Magerkohle Fettkohle Gas-Gasfl. Braun-Kohle Torf

FIG. 4 AIR REQUIREMENTS AND FLUE GASES DEVELOPED AT VARIOUS DEGREES OF EXCESS OF AIR

(*Luftbedarf und Rauchgas* = Air required and flue gas. *vH* = per cent. *vH Gehalt an flüchtigen Bestandteilen* = per cent content of volatile constituents. *Koks* = coke. *Anthrazit* = anthracite. *Magerkohle* = lean coals. *Fettkohle* = rich coals. *Gas-Gasfl.* = gas coal. *Braunkohle* = lignite. *Torf* = peat.)

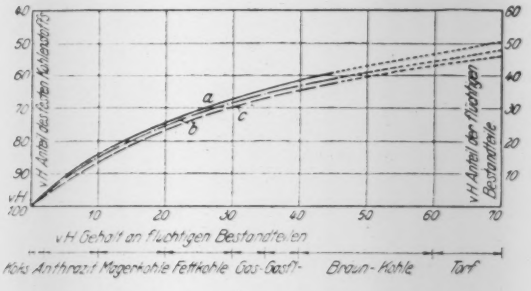
vH Anteil des festen Kohlenstoffs
vH Anteil der flüchtigen Bestandteile
vH Gehalt an flüchtigen Bestandteilen
Koks Anthrazit Magerkohle Fettkohle Gas-Gasfl. Braun-Kohle Torf

FIG. 5 PART OF VOLATILE CONSTITUENTS OF COAL AS AFFECTING AIR REQUIREMENTS, FLUE-GAS AMOUNTS, AND HEATING VALUES OF FUELS

(*vH Anteil des festen Kohlenstoffs* = per cent of fixed carbon. *vH Anteil der flüchtigen Bestandteile* = per cent of volatile constituents. *Gehalt an flüchtigen Bestandteilen vH kg. Reinkohle* = content of volatile constituents in per cent per kilogram of coal free of ash and water. *Koks* = coke. *Anthrazit* = anthracite. *Mager* = lean. *Fett* = rich. *Gas-Gasfl.* = gas coals. *Braunkohle* = lignite. *Torf* = peat.)

volatile constituents are practically the same for gas coals from peat to regular gas coal, but fall off when passing from these to coke.

The curves showing the amounts of flue gas produced and the heating values of the coals are similar to those dealing with the air requirements. Wide deviations are evident only in reference to the heating value of the volatile constituents. Similar curves are obtained (Fig. 4) in considering excess of air referred to air requirements and flue gases, but these curve more with rising excess of air. The curves for the air requirements and flue gases for the volatile constituents of coal diverge as the gas content of the coal increases, while the curves for the fixed carbon behave in the opposite way. Since this latter effect is stronger than the former, the curves of the air requirement and flue gases developed for the total coal are wider apart in the case of lean coals than in the case of gas coals, which means that with these coals an extremely great excess of air produces worse results from the point of view of firing than in the case of gas coals. The reason for this is the great demand for air on the part of the free hydrogen, which results in flue gases having a high burden of nitrogen, whereas the hydrogen already combined with oxygen produces only water vapor without nitrogen and without heating value.

How great may be the influence of the air requirements and flue-gas volume as regards volatile constituents of coal is shown by Fig. 5. For anthracite it is about 5 per cent, for lean coals 20 per cent, for rich coals 30 per cent, for lignites up to 40 per cent, and for peat

50 per cent. The influence of the oxygen content can be recognized from the flattened shape of the upper part of the curves. If the ash and water content are also taken into consideration and everything is plotted in a system of space coördinates, Fig. 6 is obtained giving a view of the values of air requirement and flue-gas production of fuels. As the ash and water contents increase the lines become flatter and flatter, so that at 25 per cent excess of air

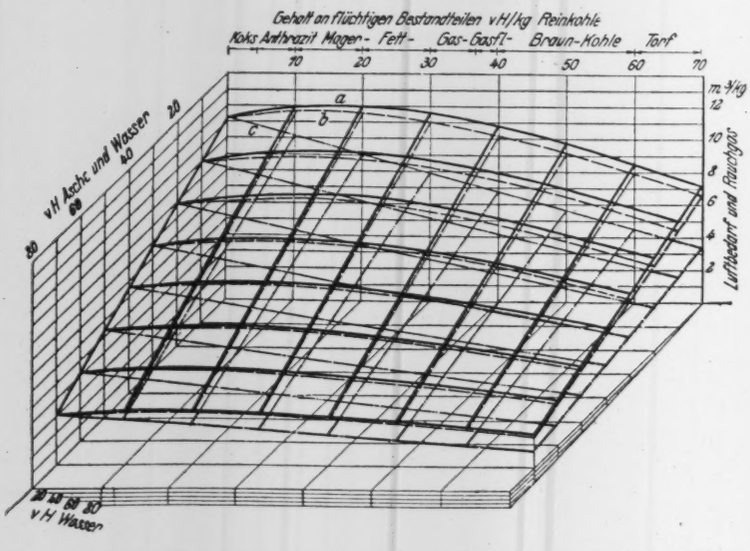

FIG. 6 AIR REQUIREMENTS AND FLUE GASES DEVELOPED PER KG. OF RAW COAL (γ = 1.25 PER CU.M.)

(*a* Rauchgasmenge = flue gas; *b* Heizwert = heating value; *c* Luftbedarf = air required; *vH Asche und Wasser* = per cent of ash and water, *Luftbedarf und Rauchgas* = air required and flue gas, *vH Wasser* = per cent water.)

and 70 per cent ash and water, the curve showing the air requirements and flue-gas mass is practically straight. The curves lying toward the region of gas coals are somewhat inclined. The equalizing influence of the ash and water content is clearly noticeable.

COMBUSTION OF VOLATILE CONSTITUENTS

The duration of burning and the perfection of combustion of the gases are influenced by the following factors: (1) velocity of burning of the gas, (2) temperature of combustion, (3) air temperature, (4) amount of excess air, (5) oxygen content of the air, (6) mixture with the air, (7) shape and dimensions of the ignition arch, and (8) shape and dimensions of the combustion chamber. As regards combustion temperature, one has to consider in the first place the temperature of the gas, because the carbon lying on the grate is not usually in full glow and hence not at its highest temperature. The temperatures of combustion of the gases contained in the solid fuels are extremely variable, depending on the content of the volatile constituents of the coal, the excess of air present, and the water content. The ash content has no influence in this connection. The curves in Fig. 7 (the original article also contains a table) give the theoretical temperatures, and from these it would appear that the combustion temperatures are highest in the case of coal containing an average of 15 to 25 per cent of gas and that these temperatures fall off regularly on both sides but decline more in the case of gas coals, being, for example, with lignites about 150 deg. cent. lower than with rich coals. As the excess of air increases, a practically uniform drop in temperature takes place. At the same time the excess of air acts as an equalizer in that the lines in the graph come closer together with increasing excess of air. The water content of the fuel is of the utmost importance. Its influence is less strongly apparent where the gas content is very great, so that lignites, notwithstanding their high water content, produce comparatively high temperatures, but in the case of fuels low in gas, lean coals, and anthracites there occurs such a sudden drop in the temperature curve that even with the moderately low water content of the coal there may be a drop below the temperature of ignition. This is the reason why watery lean fuels are so hard to ignite and so difficult to maintain in the state of combustion. It is, however, important to emphasize that the combustion temperature of the gas is not the only factor that determines the temperature of the ignition arch, the radiation from the glowing bed of coke producing also an influence in this direction. In the case of lean fuels the influence of this radiation from the coke bed is greater and this counteracts the unfavorable influence of the high water content. Attention should be called to the generally well-known fact that a certain water content favors the combustion of lean fuels. The actual cause of the phenomenon is not yet clear, but it is probably due to the catalytic influence of the water vapor.

It is impossible to achieve in the combustion chamber a good mixture of the gas with the air of combustion. It is true that the air for combustion enters the combustion chamber in a state of fine subdivision, because of its passage through the numberless interstices of the bed of fuel on the grate. On the other hand, however, it enters the combustion chamber from one side only, and since the air for combustion is heavier than the gases and since both of them fall immediately under the influence of the chimney draft, there is always present the danger of stratification within the combustion chamber in accordance with the respective specific weights. In chemical processes special constructions are employed to counteract these tendencies, but for certain reasons it is not feasible to use them in furnaces. It might be possible, however, to employ in burning coal on a grate methods similar to those used in pulverized-coal firing, namely, to produce in the combustion chamber the desired turbulence and thorough mixing by air injection. Actually, however, it is possible to employ this only to a limited extent, because

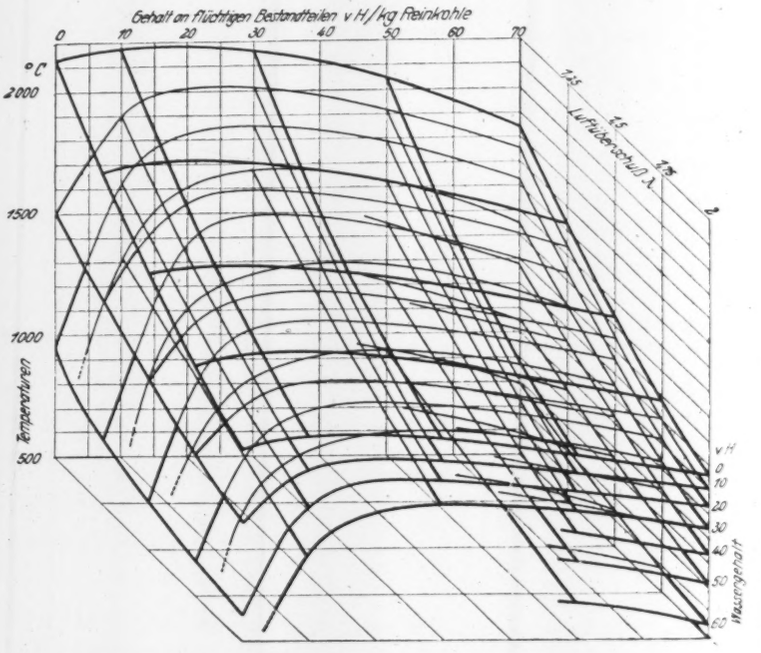

FIG. 7 TEMPERATURES OF COMBUSTION OF THE VOLATILE CONSTITUENTS OF COAL AT VARIOUS DEGREES OF EXCESS OF AIR AND VARIOUS WATER CONTENTS OF THE FUEL

(*Gehalt an flüchtigen Bestandteilen vH/kg. Reinkohle* = content of volatile constituents in per cent per kg. of coal free of ash and water, *Luftüberschuss* λ = excess of air λ, *Wassergehalt* = water content.)

as the air is introduced into the furnace from above the draft from below is reduced to the same extent, and as a result of it the velocity of combustion of the coal on the grate is correspondingly delayed.

Because of this air admission to the upper part of the firing chamber is used in connection with traveling grates and stokers only under exceptional conditions. This may be done in the first place with fuels very rich in gas, such, moreover, as are burned with a deep layer of coal, so that under certain conditions carbon monoxide

formation takes place in the fuel bed, which produces an increase in the amount of combustible gases available. This occurs, for example, with the Vesuvio grate (see MECHANICAL ENGINEERING, vol. 45, no. 11, Nov., 1923, p. 654). In order to obviate the harmful influence of the reduction of the flow of air from under the grate, one might, in the case of traveling grates, shut off the draft in the first section of the grate, i.e., the zone of degasification. Here the draft actually hinders the ignition of the coke residue. Instead of this the same amount of air preheated as highly as possible may be blown on to the grate from above. If this is done the blast will produce a powerful turbulence in the gases and will materially accelerate their combustion, this action being increased still more on account of the preheating of the air of combustion. Up to a certain point the shape of the roof may assist the mixing; this will occur if the roof is given a backward sweep so as to compress the gases. This can be used, however, only with lean and low-grade fuels. (First part of an article by Schulte, Director of the Boiler Inspection Association in Essen. *Zeitschrift des Vereines deutscher Ingenieure*, vol. 69, no. 29, July 18, 1925, pp. 941–946, 8 figs., *g*. An abstract of the second part may appear in an early issue if space is then available.)

Short Abstracts of the Month

AERONAUTICS

Present Status of the All-Metal-Airplane Industry in France

THE author deals chiefly with the methods for the construction of the all-metal airplanes used by the Bréguet Company.

Among other things, the striking statement is made that this particular company has found it necessary to organize its production in such a manner as to secure an output of one Type-XIX airplane *every three hours*. The XIX Bréguet machine is the one most extensively used by French pilots for long-distance flights and was used by Captain D'Oisy for his flight from Paris to Shanghai, China.

It is also stated that because of the high grade of organization developed, employment of improved machinery, and, where needed, highly skilled labor, the all-metal airplane does not cost much more than a wooden one and has a very much longer life than the latter.

An advantage in the manufacture of the all-metal plane as compared with the wooden plane (at least in France) is that while in the wooden plane every part has to be made by a skilled workman, unskilled labor may be used to operate many of the machines producing the metal parts. This is illustrated graphically in the original article, from which it appears that while the raw material costs more in an all-metal plane, the cost of labor is the same in both cases, although the number of hours consumed is greater in the case of the metal plane. (*Revue de l'Ingenieur*, vol. 22, no. 6, June, 1925, pp. 177–184, 10 figs., *cd*)

The Savage-Bramson Anti-Stall Gear in Practice

THE instrument consists of a stall detector and a warning device pneumatically intercoupled. The stall detector is a horizontal vane facing the air stream and unstably hinged with reference to it. It is something like the tail of an airplane, but travels hind end first. The vane is so set that just before stalling the wind catches its underside and flicks it up against the upper stop. In doing so it opens a miniature mushroom valve which, in turn, brings the warning device into operation through a pneumatic relay.

The warning device is a piston and cylinder unit. The piston is attached to a bracket clamped to the joystick about $1^1/_2$ in. below its pivot. The cylinder is attached to a cross member of the fuselage in front of the joystick. When the stall detector is pushed up a sudden forward pull is felt in the joystick; when the vane is pushed down the pull disappears.

The gear was tested in flight. The vane remained quite steady in its bottom position, but when the machine is stalled, no matter how, it is impossible to reach the stalling point without receiving the reminding tap on the stick just beforehand. The pull on the joystick is, however, only light and can be easily resisted by hand. If the machine is stalled repeatedly in quick succession there is a tendency for the warning impulse to become weaker. On making a normal "three-point" landing the gear comes into operation. This is a little disconcerting at first and in the opinion of the author of the article (said to be a well-known commercial pilot) a cut-out should be fitted so that it is left to the decision of the pilot whether or not the gear shall operate when landing. (*The Aeroplane*, vol. 29, no. 8, Aug. 19, 1925, p. 234, *d*)

AIR MACHINERY (See also Testing and Measurements: Interference Tests on a Pitot Tube)

The Wing Rotor

DATA on the so-called "wing" rotor invented by a Finnish engineer, Sigurd Savonius. Like the Flettner rotor, this one uses the so-called "Magnus effect," in accordance with which if a cylinder is rotated in an air current there is an effect tending to push the cylinder across the direction of the wind.

There are several types of wing rotors. In one of them the wings are journaled in such a manner that if the speed exceeds a certain figure the centrifugal force acts upon the wings so that they turn toward one another. Should the speed fall they are drawn from one another by compensating springs.

The Savonius rotor embodies the following principle: By the movement of the wing against the wind there is created a vacuum behind its concave back, which vacuum is said to be four times greater in effect than the pressure on its front, and acts strongly as a brake. The wings are arranged in such a manner that an opening is formed between the inner edges around the axis of rotation. With this arrangement it is claimed that the circumferential speed rises to 1.7 times the speed of the wind. A rotor of this kind was installed on a boat. In this the wings were journaled on pins in the end plates so that they could be adjusted for different speeds and strength of the wind, or reverse, so that the direction of rotation was altered for tacking. It is said that the small trial boat reached, with wind power alone and two wing rotors having a total projected surface of 3.75 sq. m. (40.35 sq. ft.), a speed of 6 knots in a wind of 10 m. (32.8 ft.) per sec. (*Mechanical World*, vol. 78, no. 2016, August 21, 1925, p. 149, *g*)

Small German Wind-Power Plants for Foreign Service

THE author believes that nature provides favorable wind-power conditions where available water power is lacking, but the uneven supply of wind power available throughout the daily and yearly periods makes it necessary to provide in connection with wind-power plants a generously dimensioned reservoir of energy, such as a storage battery.

A wind-power installation will pay for itself when wind is available at a velocity of 6 to 7 meters (20 to 23 ft.) per sec. for at least 2000 hr. per year. There are two main constructions of wind-power units, one of the heavy type with a dynamo under the roof and driven by a transmission shaft with gears, and a light construction where the dynamo is coupled to the air motor directly or through a simple drive and is located at the top of the wind tower. The former construction cannot be used for outputs of less than 3 kw. nominal, and is used where the low speed of the wind wheel makes it imperative.

The author claims that, in general, over the 24-hr. period the wind velocity is under the daily average for 15 hr. and above the daily average for 9 hr. The maximum wind velocity occurs most often in the early afternoon and a second maximum in the late afternoon. Then toward night the wind may die out entirely. In mountainous regions and on the seashore these variations occur likewise, but in a more pronounced form. The author cites five cities in four countries of Europe to show how the wind varies from a maximum to a minimum over the daily period, and finds that the variations are the same on the German coast, in Vienna, Austria, St. Petersburg, Russia, etc.

Attention is called to the fact that the few meters of elevation of the wind-motor tower produce an appreciable increase in the wind velocity or output. For heights in excess of 16 meters (52.5

ft.) above the level of water, the wind velocity increases with the fifth root of the height of the tower, and the output increases with the 0.6 power of the height of the tower. Actual measurements of a 15-meter (49.20-ft.) mast have given the following values (1 meter = 3.28 ft.).

Height of mast, meters.....	0	3	6	9	12	15.2
Wind velocity, meters per sec...	3.6	8.2	8.7	9	9.1	9.4

As regards the yearly variation, the article cites data obtained on the Eiffel tower in Paris. Other similar data are cited.

When wind motors are to be installed in mountainous sections or at high elevations, the height above sea level should be taken into consideration. Since the wind pressure is proportional to the density of air, which varies with the elevation above sea level, the output of the motor decreases as compared with sea-level conditions. Thus at 15 deg. cent. (59 deg. fahr.) and elevation of 1000 meters (3280 ft.) the output is 11 per cent less than on sea level. Where, however, the local conditions have been properly taken into consideration, wind-power installations give excellent satisfaction, as shown by a number of units which have been running in Germany for as long as 15 years.

Among other things, the article gives data, some of which are not original, as to wind velocities in Argentina and Brazil. The article gives also formulas for determining the output of an air motor as a function of wind velocity and area of the wheel. As regards the distribution of costs of a wind-power installation, it is stated that the mechanical part comprising the wind turbine proper takes 35 per cent of the overall cost; the generator with switches and wiring, 15 per cent; the storage battery, 35 per cent; building and freight of the material to destination, 15 per cent. (Kurt Lubowsky, Doctor of Engineering, Berlin, in *Elektrotechnische Zeitschrift*, vol. 46, no. 26, June 25, 1925, pp. 949–953, 4 figs., *dg*)

A Method of Determining the Dewpoints of Fuel-Air Mixtures

A VERY important consideration limiting the proportions of low-volatility constituents permissible in a hydrocarbon engine fuel is the minimum temperature required to keep a mixture of the fuel and air completely vaporized. This temperature is called the dewpoint of the mixture. This paper outlines a method which has been devised at the Bureau of Standards for determining these temperatures for mixtures of any fuel with air by means of a few measurements on the fuel and on the mixture. The method has been checked by applying it to a variety of gasoline-air mixtures and comparing the results with those of a more direct method.

The theory is based on the assumption that over the small range of pressures and temperatures with which we are here concerned (those occurring in the engine manifold), the first very small fraction of liquid condensing out just below the dewpoint and in equilibrium with the remaining vapor is of sensibly constant composition. The Clausius-Clapeyron equation relating the rate of change of vapor pressure with temperature to the heat of vaporization of the first condensate is then applicable, and by means of this relation, the equation of condition for ideal gases, and Dalton's law a closely approximate expression is derived for the dewpoint corresponding to any pressure and mixture ratio. It is

$$T = \frac{K_1}{\log P - \log\left(1 + \frac{M_t}{M_a} R'\right) - K_2}$$

where T is the dewpoint; K_1 and K_2 are experimentally determined constants characteristic of the fuel; M_t and M_a are the average molecular weights of the fuel and of air; R' is the mixture ratio; that is, the ratio by weight of air and fuel; P is the total pressure exerted by the fuel-air mixture.

The average molecular weights are determined by any of the classic methods, which consist essentially in measuring the quantities, p, v, m, and T (the gas constant R being known), in the equation of condition

$$pv = \frac{m}{M} RT$$

and from them computing M.

To determine the constants K_1 and K_2, a small quantity of liquid fuel is accurately measured into a partially evacuated glass vapor chamber immersed in a variable-temperature bath, and the temperature is noted at which condensation just appears or disappears on a small spot which is kept a fraction of a degree cooler than the rest of the chamber wall while the bath temperature is decreased or increased. The average molecular weight of the solution having been determined, the partial pressure corresponding to any temperature can be computed. If logarithms of partial pressures are plotted against reciprocals of the corresponding initial absolute temperatures, a very nearly straight line is obtained; two such pairs of values are theoretically sufficient to fix it. From the derivation of the equation for T, K_1 is seen to be the slope of this line, and K_2 is determined by its intercepts.

With the numerical values of the four constants substituted in the equation, it expresses the dewpoint as a function of the pressure and mixture ratio of the mixture. Curves are plotted between dew point and pressure for several different values of mixture ratio for each of four gasolines investigated. (Roy J. Kennedy. Abstract of *Scientific Papers of the Bureau of Standards*, no. 500, *e*)

ENGINEERING MATERIALS

Testing of Fireclay Brick with Special Reference to Their Use in Coal-Fired Boiler Settings

FOLLOWING a conference of Government representatives and users and producers of fireclay refractories, held at the Bureau of Standards, a voluminous field-survey report was obtained through the coöperation of Stone & Webster, and refractories of 42 brands, representative of the product as manufactured in the United States, were submitted to an extended investigation in the laboratory.

In the laboratory the refractories were subjected to the following tests: (1) An endurance test in which brick were held at 1450 deg. cent. for 72 hr., both with and without load; (2) a reheating test in which the change in volume and porosity were determined after the brick had been held at 1400 deg. cent. for five hours; (3) a quenching test; (4) the standard A.S.T.M. load test, and two modifications of this test; and (5) the softening points were determined by the cone method, and brick of each brand were analyzed chemically and several examined petrographically.

As a result of the laboratory work, it was found that a close relation existed between data obtained in the endurance, reheating, quenching, and softening-point tests, and that these depended to a remarkable extent on the chemical composition. The results indicated that a refractory which would successfully withstand 15 quenchings from 850 deg. cent. to running water had a softening point equivalent to at least that of cone 32; that the linear change in the endurance test would not exceed 2 per cent and when transversely loaded the deflection would not exceed $^{10}/_{32}$ in.; that the percentage absorption after the reheating test would lie between 6 and 10 per cent; and also that the refractory should be composed of raw materials containing not more than 4 per cent flux and 20 per cent uncombined quartz. The results also favor the dry-press process for manufacturing brick.

It was found that refractories which appeared of highest quality in the laboratory did not always give the highest service, but this was to be expected since no two brands of refractories were subjected to similar service conditions. The field survey showed, however, that in practically every instance brick fail primarily through erosion, indicating lack of refractoriness and resistance to the fluxing action of molten coal ash. The one exception was the suspended arch in which the refractories failed mainly by spalling. In addition, the data from the survey indicated that where the life was relatively long the brick failed by erosion while, with a short life, the failure was due to combined erosion and spalling. (R. F. Geller. Abstract of *Technologic Paper of the Bureau of Standards*, no. 279, *e*)

FOUNDRY

Castings Strengthened by Addition of Nichrome

IT WAS found at the works of the International Motor Co., maker of Mack trucks, that improved and stronger iron castings can be produced by adding to the metal about 2 per cent of ni-

chrome, a patented alloy manufactured by Driver-Harris Company. The metal is added to a hot ladle which is then filled with molten iron from the cupola. The nichrome melts almost instantly and appears to be evenly distributed throughout the ladle. It produces a stronger casting and one having the grains smaller and the graphite flakes finer and more widely dispersed. Castings of 230 Brinell hardness made of this kind of iron machine with no more difficulty than ordinary gray iron of 170 Brinell hardness. It is also said that there is no appreciable difference in hardness or grain structure throughout the casting. Castings of this kind of iron are to be used for cylinder blocks, pistons, differential housings, and the like.

It is also stated that since 1912 the cost of repairs required by Mack trucks has been reduced over 75 per cent. William Day, Jr., metallurgist of the International Motor Co., claims that much of the credit for this is due to the use of nickel alloy steel for such parts as jackshaft, axles, frame, steering knuckle, etc. (*Automotive Industries*, vol. 53, no. 8, August 20, 1925, pp. 295–296, *d*)

FUELS AND FIRING

Bergius Process of Fuel Treatment

GENERAL data on the Bergius process were given in MECHANICAL ENGINEERING, vol. 46, no. 4, April, 1924, p. 215. By this process with the additional use of hydrogen a large share of the weight of the coal is converted into a liquid fuel. It is now stated that the German State has given a grant or guarantee of two and a half million marks toward the cost of erecting works at the Wenceslaus coal mines in Lower Silesia for the treatment of large quantities of coal by this new method.

It is stated that it does not follow from this that the commercial success of the process has already been established. British coals appear to be especially suitable in composition for treatment by the Bergius process, and at Mannheim such coals have been transformed into liquid and gaseous products to the extent of 85 per cent by weight at the expense of hydrogen equal to 10 per cent by weight of the coal. The liquefaction process may make available for fuel large quantities of "Duff" coal, of which little if any use is now made. The Bergius process is said to be controlled by the Badische Company. (The *Chemical Age*, vol. 13, no. 322, August 15, 1925, p. 165–166, *g*)

The Iron and Coal Trades Review (vol. 111, no. 2998, August 14, 1925, p. 251) states that it is currently reported that the Badische Company has succeeded in producing synthetic motor fuel from carbon monoxide by a catalytic process. [The same company controls the synthetic production of methanol which attracted such wide attention recently in America.]

INTERNAL-COMBUSTION ENGINEERING

New Single-Acting and Double-Acting Krupp Diesel Engines

THE single-acting engine develops 1600 shaft hp. in four cylinders. The cylinder head is of normal design except that the fuel valve is offset from the center line of the cylinder. Below the cylinder is a water-cooled shield, which is only attached to the cylinder by the water connections. There is some doubt as to the desirability of such an arrangement.

An unusual arrangement of scavenger-pump drive is found on this engine, where an arm on the crosshead serves both to carry the piston cooling-water pipes and the piston rod of a double-acting scavenger pump. It might appear at first sight that this would have a tilting effect on the crosshead, but it is claimed that on the contrary the load due to the scavenger pump merely serves to partially counterbalance the tipping effect of the connecting rod, and that the pressure at the edge of the slipper is at least 14 per cent less than it would be without this form of scavenger drive. The slight increase in the vertically moving masses is of no consequence as it only amounts to 4 per cent of the total.

For an engine of this size the air starting arrangements are unusual. Starting air and fuel are admitted simultaneously and to all four cylinders at once, with the result that a very large mean torque is obtained. It is claimed that whereas using compressed air alone the starting mean effort at right angles to the crank arm, referred to piston area, is 151 lb. per sq. in., with simultaneous fuel injection it is 180 lb. per sq. in. When the normal method of air starting is used, that is to say, when half the cylinders are put on fuel while the remainder are still operating on air, the figure is only 54.5 lb. per sq. in. under the worst condition when the cylinders on fuel have not yet started to fire.

The double-acting engine follows the design of the single-acting one fairly closely. Both are illustrated by general diagrams in the original article. (*Motorship*, vol. 10, no. 7, July, 1925, p. 528, 3 figs., *d*)

The Whaley Constant-Pressure Oil Engine

FIRST published data on the engine invented by W. H. Smith Whaley and built by the Sun Shipbuilding and Drydock Co. The engine now built is said to be a single-acting, four-cylinder, two-cycle machine rated at 750 hp. The cylinder is kept floating on a pressure reservoir both by an overflow valve and by a piston valve as long as fuel is being injected and burned.

The engine is distinguished from other heavy-oil engines by the fact that the cylinder clearance at the time of the fuel injection is opened by a piston valve communicating with the static receiver having many times the capacity of the cylinder clearance, the piston valve staying open as long as the fuel is being injected.

Merely floating the reservoir on the combustion space is said to require little actual discharge or return through the valves; at the same time it is hardly to be expected that enough fuel could be burned within the limited confines of the combustion space to cause a substantial rise of pressure in the system. By means of piston valves the same combustion-space port is made to serve successively for the introduction of the super-scavenging "kick" and to establish communication with the reservoir during the combustion period. Devices arranged on the piston valve stem actuate pumps spraying fuel by direct injection at 6000 lb. per sq. in. As it is recalled that the valve is linked up by means of the double eccentrics and that its travel is thereby altered, the possibility for maintaining the relation between the fuel-injection process and the duration of the valve opening to the reservoir is established and rendered subject to operating control.

Stress is laid on the fact that practically no rise in pressure beyond that due to compression is expected. In practically all oil engines operating with a trapped air charge confined by a closed combustion space, at least a 10 per cent rise in pressure due to combustion is essential to the attainment of economical running—not so much, probably, because of the extra force exerted on the piston as on account of the earlier timing of fuel injection and the progress of combustion at a higher temperature. In the Whaley engine even early timing would probably cause no substantial pressure rise up to the capacity of the transfer valves.

Holding maximum pressures to a definite figure thus becomes the means of accurately carrying out a program for the design of a lightweight engine. As the factor of safety is more closely known when variations in maximum pressures liable to occur under service conditions are practically eliminated, both the framework and moving parts of the engine can be proportioned accordingly. It is claimed that the 750-hp. engine recently exhibited weighs only 68,000 lb., or about 90 lb. per hp., a figure which would represent a substantial improvement over current practice. For marine work such a reduction in weight would have obvious advantages, and as manufacturing costs are often found to be proportional to the weight of the engine, there is a considerable economic advantage to be gained also for land engines by the elimination of superfluous metal.

Another feature which is said to aid in reducing weight per horsepower is the uniflow scavenging obtainable by allowing the "kick" of high-pressure air to pass through the combustion space and out through the ports. The valve through which this air enters is opened only a few degrees before the closing of the exhaust ports and remains open long enough after closure for the cylinder to be filled with air having a considerably higher density than that of the atmosphere. The horsepower rating of the engine would be directly increased by the extra weight of air present in the cylinders.

As compression therefore begins with, say, 45 lb., a relatively small reduction in volume suffices to bring it up to the 350 lb. corresponding to the temperature required for ignition. That means that the combustion or clearance space may be larger and may lend itself more easily to a favorable location of the fuel spray and other valves than is the case with engines beginning to compress at

about atmospheric pressure. (*Oil Engine Power*, vol. 3, no. 9, September, 1925, p. 509, *d*)

Recent Oil-Engine Developments

A REVIEW of the British oil-engine industry listing the types in actual production and the more important experimental work. Only data referring to types not generally known in this country will be reported here.

Among full-Diesel engines the following may be mentioned particularly.

The interesting feature of the Parsons engine lies principally in the general layout of the plant and the incorporation of a high-speed type of engine. The engines are of the conventional trunk-piston or crosshead type, according to the horsepower employed. The trunk-piston design is retained for the smaller units developing 300 shaft hp. in 6 cylinders at a speed of 450 r.p.m. The crosshead construction is used for units (also 6 cylinders) developing 600 shaft hp. at 300 r.p.m. Both types incorporate an enclosed crankcase, forced lubrication, and a four-stroke cycle of operation. The two proposals so far suggested by the makers deal with an installation of 1200 shaft hp. total; hence four or two engines, respectively, are grouped together fore ach plant. The drive from each engine is taken to the propeller through a single reduction-gear arrangement with a final propeller speed of 75 r.p.m.; hence both engines and propeller are run at efficient speeds. The flywheel is constructed as a friction clutch, and an elastic frictional coupling connects the engine crankshaft to the pinion of the reduction gear. In the case of a four-unit installation the four pinions are grouped around the single reduction wheel mounted on the propeller shaft. For that reason the four engines are arranged in two tiers. Another feature is the utilization of the heat from the jackets and exhaust gases. All cylinder jackets are interconnected and form a closed circuit with an oil-fired boiler. Interposed in the circuit are vertical regenerators through which the exhaust gases pass, and thus give up further heat to the circulating water. The whole system is constructed to deal with steam up to 250 lb. per sq. in. Steam is generated in the boiler, circulation commences, and the cylinder jackets become warmed up.

Steam is also used for starting-up purposes in place of compressed air, the fuel valve and steam starting valve being interconnected in a very similar manner to that used with conventional air starting devices. The preheating of cylinder jackets and liners and the absence of the chilling effect when starting up on steam, enable the introduction of a lower compression pressure of about 350 lb. per sq. in. Also the hot conditions of the combustion space facilitate a quick start with a resultant low steam consumption. When at sea sufficient steam is generated without the aid of the oil-firing equipment to run the auxiliaries; hence a considerable saving of fuel is introduced.

The compactness of such a scheme is indicated by the engine rooms' being only 26 ft. 9 in. long plus a 5 ft. 6-in. recess when dealing with a four-unit installation. The total weight of machinery, including reduction gearing and propeller gear, is 225 tons. By replacing the four units by two sets, the engine-room length is increased to 32 ft. plus 5 ft. 6 in. recess, and the total weight amounts to 266 tons. The system is interesting from the point of view of utilizing the heat from circulating water and exhaust gases, and the entire elimination of compressed air.

As regards the Werkspoor, of particular interest is an eight-cylinder, 2200 shaft hp. unit built by the North Eastern Marine Engineering Co., Ltd. This engine represents the highest-powered engine of the Werkspoor class so far completed. Incidentally cast-iron columns have been introduced. The cylinder dimensions are $28^3/_4$ in. diameter by $51^1/_4$ in. stroke, using a compression pressure of 465 lb. per sq. in. Full power is developed at 92 r.p.m. with a m.i.p. of 95 lb. per sq. in. The mechanical efficiency at full load amounts to 75 per cent; hence the total i.hp. is 2930. For full load the fuel consumption on the test bed amounted to 0.404 lb. per shaft hp-hr.

The Vickers was one of the earliest exponents of the mechanical injection system in England. While until recently the largest size of engine built by this firm developed 1250 shaft hp. (24.5 in. diameter by 39 in. stroke at 118 r.p.m.) in 6 cylinders, there is now completed a 2700-shaft hp. set developing full power at 110 r.p.m. with 8 cylinders (30-in. diameter by 45-in. stroke). This is the highest-powered single-acting four-stroke engine which so far has been completed.

In the Doxford a comparatively low compression pressure of 300 lb. per sq. in. is used, and to facilitate starting, steam is circulated to warm up both cylinder jackets and pistons. Combustion is thus completed under favorable conditions and remains quite positive at such low speeds as 16 r.p.m. So far 745 shaft hp. or 883 i.hp. have been developed in one cylinder with a 22.8-in. bore and 2 × 45.6-in. stroke, the normal speed being 90 r.p.m. The total weight of a three-cylinder engine of this size is 290 tons, and a fuel consumption of 0.37 lb. per shaft hp-hr. has been recorded. The mechanical efficiency is 85 per cent, and of the mechanical losses 10 per cent is absorbed by the engine, $3^1/_2$ per cent is required for the scavenging pump, and $1^1/_2$ per cent for lubricating and cooling pumps. The lubricating-oil consumption for this size amounts to $9^3/_4$ gal. per day of 24 hr.

Of the Fullagar, the largest engine so far constructed is a 2700-shaft hp., six-cylinder unit, 23 in. bore and 36 in. stroke, making the total effective stroke 72 in. The engine weighs 359 tons. With a compression pressure of 420 lb. per sq. in. (550 lb. maximum pressure), a m.i.p. of 87 lb. per sq. in., in full power, is developed at 86 r.p.m. The fuel consumption is 0.38 lb. per shaft hp-hr., and the lubricating-oil consumption amounts to 12 gal. per day of 24 hr. The mechanical efficiency is 80 per cent, the losses being made up as follows: air compressor, 6 per cent; scavenging pump, 4 per cent; lubricating and cooling pumps, 2 per cent; and other losses (frictional, etc.), 8 per cent. As in the case of the Doxford engine, the slow running speed is in the neighborhood of 16 r.p.m. equivalent to a piston speed of 96 ft. per min.

Of the semi-Diesel engines the Mumford and White deserve particular mention. Although the Mumford engine is not yet on the market, the particulars so far available are very interesting. The design is in general appearance that of a high-speed vertical type four-stroke paraffin engine. The atomization of the fuel is effected by a vaporizer which is of a type to allow a sufficiently wide speed range to accommodate the requirements of a high-speed marine engine. The engine so far built is a four-cylinder unit developing 74 shaft hp. at 900 r.p.m., with a cylinder bore of $5^1/_2$ in. and a stroke of 9 in. The piston speed is 1350 ft. per min., and since as high a speed as 1400 r.p.m. has been recorded, the design incorporates piston speeds up to 2100 ft. per min. At the other end of the range a speed of 300 r.p.m. has been maintained for periods to satisfy practical requirements.

As this type of engine has considerable scope for both water and road transport further developments will be awaited with interest, particularly since this make represents one of the very few examples of high-speed constructions that are at present being experimented with in Great Britain. As already pointed out, much more activity is displayed on the Continent with regard to this class of engine.

The White engine is a somewhat novel departure from conventional two-stroke crankcase-compression design. Much has recently been said with regard to supercharging applied to oil engines, and in the White engine we have what is probably the first practical example incorporating a system of increasing the power rating by pumping additional air into the cylinder to raise the output per square inch of piston area. Crankcase compression is retained, but to increase the air in the cylinder a double-acting air pump with a capacity of 120 to 150 per cent of the volume of the power cylinder augments the air supply to the cylinder. In the two-cylinder construction with cylinder bore 12.8 in. and stroke 14.8 in., a m.i.p. of 60 lb. per sq. in. is introduced for normal rating of 120 shaft hp. at 275 r.p.m. The cylinder head is completely water-cooled and a mechanical-injection type spray valve with a compression pressure of about 250 lb. per sq. in. accounts for the effective combustion of the fuel. The admission of the supercharging air through a special series of ports is controlled by a rotary valve fitted in suitable casings attached to the cylinder jackets. The flexibility in load and speed range is secured by regulating the amount of supercharged air so that the small uncooled hot plate in the cylinder head retains the necessary temperature at low load and speed conditions. In other respects the engine incorporates constructional details nowadays standard practice for this class of engine.

The hot-bulb engines are principally of the two-stroke crank-

case-compression type and are considered as wasteful in lubricating oil, and partly so in fuel. The author believes that to avoid this condition the only effective means is to eliminate crankcase compression by using separate scavenging pumps, although it is found that the cost of the engine so equipped will be higher. The new Crossley incorporates such a separate scavenging system. The scavenging pump is mounted on a separate crank built as an extension of the main crankshaft. The cylinder head is completely water-jacketed, and the fuel nozzle of the mechanical-injection type, together with a compression pressure of 270 lb. per sq. in., accounts for the ignition and combustion of the charge. The engine, built so far as a standard size, develops 14 b.hp. (6-in. bore by 8-in. stroke) per cylinder at a speed of 450 r.p.m. and works on the two-stroke cycle. For starting up a half-compression cock is used, which discharges into the exhaust manifold for purposes of silent running. The engine uses a m.e.p. of 55 lb. per sq. in., which is considerably higher than in the more orthodox types.

In the Mirrlees engine, also a newcomer to the oil-engine market, the scavenging air is produced in a chamber in which the underside of the working piston acts as the compressor piston. The lower end of the cylinder is closed by a cover carrying a stuffing-box for the piston. The air passes from this chamber through a duct to the cylinder. The air pressure on the crankshaft bearing is eliminated; contamination of lubricating oil and fuel oil is avoided; heavy consumption of lubricating oil through being drawn into the cylinder is prevented. The A-frame has been introduced together with a crosshead construction, and as a result a rather higher engine has resulted. The principal consideration is capital costs, and despite a separate liner and such other manufacturing refinements, the engine is little more expensive than the standard crankcase-compression design.

Mechanical injection as the outstanding feature of the cold-starting type is said to be making rapid progress, and many Diesel-engine manufacturers are experimenting with the view to eliminating the air compressor.

A new type of "spring injection" has been introduced in the Blackstone engine (see MECHANICAL ENGINEERING, vol. 42, no. 8, Aug., 1920, p. 466). The essential parts of this new system comprise a fuel-measuring pump, a high-pressure fuel-injection pump and a spring-loaded fuel-injection valve, with a multi-orifice spray nozzle. The fuel measuring pump with a plunger for each cylinder delivers the correct amount (measured out by a wedge under control of the governor) to a second pump barrel in the spray box attached to the back of a completely water-cooled cylinder cover. There is thus no high-pressure line between the fuel pump proper and the valve box in the cylinder head. The fuel, on entering the spray box, displaces a composite plunger, of which the outer member is spring-loaded. On the return stroke of the spring plunger (mechanically operated movement) the oil charge is compressed and the resultant pressure acts on the spring-loaded spray valve. This valve is kept on its seat not only by a spring but also by a stop which is connected by a lever to the operating gear of the spring plunger which is compressed during the delivery stroke. The stop release motion is timed and hence the fuel is injected always at the same point of the engine stroke irrespective of the load. It will be noted that the pressure of injection is always constant irrespective of the speed of the engine. Hence the spray velocity is constant and proportionate to the load on the engine. As a general result of the injection system the combustion takes place partly at constant volume and partly at constant pressure, thus giving a high combustion efficiency. A turbulence block is attached to the top of the piston and a number of refinements are introduced, e.g., half-compression gear for starting; spring-loaded relief valve to fuel-measuring pump; cylinder relief valve preventing high cylinder pressures, etc.

In the Tangyes cold-starting engine automatic deflection of the fuel spray in the direction of a hot spot is used when starting up from cold and while gathering speed. The pump pressure is then not at its maximum intensity, which causes a similar lift of the fuel valve, and the shape of the fuel valve allows the fuel spray to impinge on the hot spot situated in the side of the cylinder head.

In the discussion of the paper which followed, J. Hamilton Gibson expressed his belief that experiments were being made with steam injection with fairly good results.

H. Bentham suggested that some improvements in efficiency might be obtained if the intake air was passed through an air-jacketed exhaust pipe and heated to as high a temperature as possible before admission to the cylinder. Expansion of air might be counteracted by some simple arrangement for compressing the air to the required density; at the same time the engine would run more smoothly and probably greater flexibility would be obtained by being able to feed air into the cylinders at varying densities. (J. L. Chaloner. Paper read at the meeting of the Diesel Engine Users' Association, April 17, 1925. Original paper pp. 1–28 and numerous illustrations; discussion, pp. 38–48, also illustrated, *d*.A)

Superchargers

THE author begins by showing that for high-altitude flight a supercharger is economical from the point of view of weight. There are several classes of superchargers, of which the most successful seems to be the one driven by an exhaust-gas turbine. Other types may, however, be used, in particular one driven by an auxiliary engine or by gearing.

Either a geared or turbo supercharger used with a Liberty engine to maintain sea-level pressure at 20,000 ft. requires approximately 56 hp. The power required to drive the supercharger is partially recovered as the engine has its carburetor air supplied at 14.17 lb. per in. pressure and exhausts at only 7 lb.

The Engineering Division at McCook Field after five or six years' steady development and experimental work on the turbo supercharger has overcome practically all former difficulites. Instead of as formerly being in the nose of the machine, the supercharger is now mounted on the side of the airplane with the turbine wheel overhung and exposed to the slipstream of the propeller. In this way the buckets are kept cool, the exhaust gases discharge freely to the atmosphere without unnecessary back pressure, and the operating conditions are more favorable. The bearings have been simplified and at present it is very unusual to have any bearing trouble in spite of the fact that speeds as high as 34,000 r.p.m. have been maintained. Developments are now under way to replace plain babbitted bearings of the turbo supercharger by ball and roller bearings which will be packed with lubricating fluid and will remove the necessity for pressure oil feed to the supercharger and drain lines from the bearings.

The engine at first gave trouble when used with the supercharger chiefly because the air supplied it was hotter. This was partly overcome by the addition of an air cooler located between the supercharger and the carburetors and also by using spark plugs with larger electrodes and adequate cooling surface.

The fuel systems also gave trouble until a system was developed consisting of a geared pump driven by a flexible shaft and located lower than the fuel tank. Fuel flows to the pump by gravity and is pumped through a supercharger relief valve to the carburetors, the valve keeping a constant differential between fuel in the feed line and in the carburetor float bowl.

Certain special devices were also introduced; for example, an automatic control which will maintain sea-level pressure at any altitude without attention on the pilot's part. Next is a differential fuel-pressure gage which shows the difference in pressure between the fuel in the feed line and in the carburetor float chamber. Next is a pressure gage which shows the supercharger air pressure either in pounds per square inch or in feet altitude.

The application of the supercharged engine to airplanes increased the performance which it was possible to obtain. For pursuit work the first installation was made on an Orenco having a 300-hp. Hispano engine. The ceiling of the ship unsupercharged was approximately 22,000 ft., and with the supercharger somewhat better than 33,000 ft. The ceiling of the Martin bomber without load has been raised from 14,000 ft. to approximately 28,000 ft. by means of the supercharger, while the ceiling with full military load has been increased from 7000 to better than 12,000 ft. A DH4 with full military load and supercharger will reach an altitude of 28,000 ft. against approximately 18,000 ft. unsupercharged. It should be noted, however, that all of these installations have been made on existing airplanes, and that, with one exception, no airplanes have been designed to take full advantage of the supercharged engine. It is therefore reasonable to assume that even better results will be

obtained when this is done. The one exception the Kerber-Bouleton CO5, which was designed around the supercharged engine, successfully broke the world's altitude records, carrying weights of 250, 500, and 1000 kg. At the present time all new aviation engines are being designed to accommodate either a gear-driven or turbo supercharger and practically all engines that are used in the Air Service from now on will be equipped with some type of supercharger. (David Gregg, Research Engineer, Engineering Division, McCook Field, in *Aviation*, vol. 19, no. 4, July 27, 1925, pp. 90–92, 3 figs., *d*)

MACHINE TOOLS

Spiral Bevel-Gear Planing

THE production of spiral bevel-gear wheels and pinions by a planing process involves the reciprocation of a planing tool in a straight line toward and away from the conicala pex of the gear blank, and the simultaneous rotation of the blank continuously in one direction or the other, according as the teeth are to be spiraled to the right or the left. The process cannot be said to generate the teeth. While the spiral trend may be truly described as generated, the involute or other cross-section of the teeth has to be produced by some form of cam or similar motion.

The process, although simple in its general principle, presents some complexity in its practical execution. It has the merit, however, of yielding decidedly more accurate results than the pure milling process, its common alternative.

The article next describes the Rath spiral bevel-gear planing machine manufactured by Alfred Herbert, Ltd., of Coventry, England. The principle on which the machine operates is evident from Fig. 1.

If a sharp-pointed tool is moved at uniform speed from A to D, Fig. 1, across the face of a parallel-sided blank $ABCD$, rotating uniformly in the direction indicated, the trace of the tool on the developed surface of the blank will be a straight line, such as AF. If the blank is conical, as $ABED$, its peripheral speed diminishes progressively from the large to the small end, with the result that the trace of the tool point is not straight but curved, as shown at AG on the developed surface. In practice, it is not possible to reverse the motion of the tool exactly at the point G. If we allow it to overshoot the work to some point H, and then cause it to return idly at, say, twice the speed of the cutting stroke, the course followed by the tool point will be along the reversed curve HJ. With an overshoot to K the tool begins its second cutting stroke KL, and leaves the trace MN on the work.

The fact that the trace AG is curved and not straight is not of much practical importance so far as the running of the finished pinion is concerned. It is, however, of the utmost importance that the trace on the mating wheel $PEBQ$ should be of similar curvature and slope. This result, it is evident, will be completely secured if, the speed of the tool on the cutting stroke being unaltered, the peripheral speeds of the pinion and wheel while they are being cut are equal at all corresponding points on the coned surfaces. In other words, the traces will be identical if the rotary speeds of the pinion and wheel when they are being cut are in the same ratio as the speed with which the two gears will run together when finished. This requirement clearly necessitates the provision of a considerable range of work-spindle speeds, if the capacity of the machine as regards the number of gear ratios which it will cut is not to be unduly restricted. It is to be noted that the capacity in this respect with a given number of work-spindle speeds will be multiplied if a range of speeds is provided for the tool. In general, however, the tool speed has to be chosen to suit requirements other than the attainment of a given spiral form of the traces or teeth.

If the overshoots GH and JK were zero, and if the return stroke HK were effected at infinite speed, the beginning M of the second trace would lie radially below the end G of the first. The fact that the tool overshoots the work, and the fact that the return stroke is effected at finite speed, both act to displace the point M to the right of the radial position. In the practical working of the pinion, however, continuity of tooth action requires that the point M should be to the left of the radial position, in order that each tooth may overlap its immediate successor. It is clear that such overlap as between the traces AG and MN cannot be secured unless the work is stopped and rotated backward through a certain amount during the period occupied by the overshooting and return movements of the tool. Such interruption in the motion of the work can be avoided if the trace MN be regarded not as the second, but as the third one. Thus, let AM, or GN, be made equal to twice the desired pitch distance. Then, on the first round of the blank, all the odd-numbered traces AG, MN, etc., will be cut. If twice the pitch distance goes evenly into the circumference of the blank, the tool will register with the point A at the end of the first round. Feed might then be applied to it, and the odd-numbered traces on the second and succeeding rounds cut deeper. When these traces have been cut to the finished depth, the tool would be withdrawn, and the work indexed around through one pitch distance in order that the cutting of the even-numbered traces, RS, etc., might be begun and completed.

If, on the other hand, the wheel to be cut has an odd number of teeth, twice the pitch distance will go into the circumference an integral number of times, and one pitch distance over. The tool at the end of the first round will therefore not register with the point A, but shortly afterward will start at R to cut the first of the

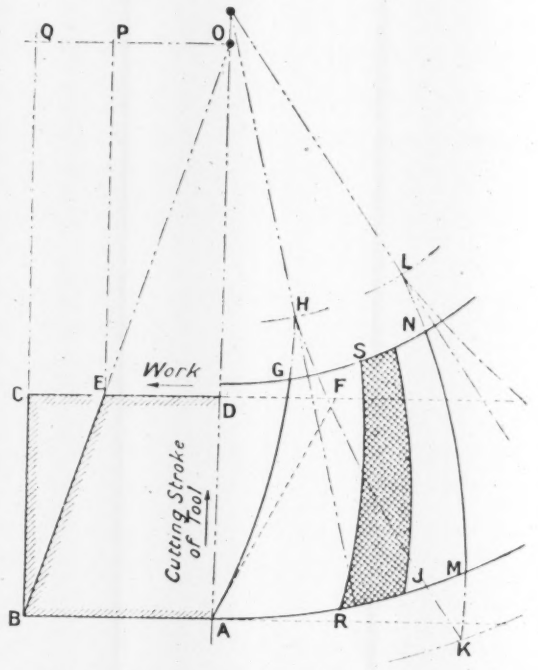

FIG. 1 DIAGRAM ILLUSTRATING OPERATION OF THE RATH SPIRAL BEVEL-GEAR PLANING MACHINE

even-numbered traces RS. Feed in this case would be applied at the end of every other round of the work, and the traces would be cut without the necessity of indexing the work at half time, as in the case of a wheel with an even number of teeth. Actually, however, there is little loss of time in the stroke from about $\frac{1}{2}$ in. to about $1\frac{9}{16}$ in. in nine steps. We further find that the speed of the tool on the idle return stroke is about three times as great as the speed on the cutting stroke. The time spent on the cutting stroke is thus approximately either 1.5, 0.9, 0.55, or 0.37 sec. The cutting speeds therefore range from about 2 ft. per min. to about 21 ft. in thirty-six steps, not all of which, however, may be different. In actual practice, this range of cutting speeds cannot be freely selected from, as will presently be made clear.

Finally, one has to consider the factors affecting the pitch distance and the number of the teeth cut on the blank. The double pitch distance AM, Fig. 1, is the peripheral length turned out by the work in the interval of time occupied by the tool in making one complete reciprocation; that is to say, in traversing the relative path $AGHJKM$. If, then, the work is rotating at n revolutions per minute and the tool is making m strokes in the same time, the angle turned through by the work while the tool is making one complete reciprocation will be $2\pi n/m$. The double pitch distance AM will therefore be $\pi dn/m$, and the true single pitch will be half this amount. Dividing the circumference of the large end by the single pitch, we get $2m/n$ as the number of teeth.

The three variable factors so far dealt with which are under the control of the operator are (*a*) the length of stroke of the tool, (*b*) the number of strokes per minute made by the tool, and (*c*) the speed in r.p.m. of the work. In the machining of a given pinion blank, the width of the blank face sets a minimum value to the length of stroke of the tool. Let us suppose the stroke length is chosen at something in excess of this minimum. With this factor tentatively fixed, the operator next selects the number of strokes per minute to give a cutting speed suitable for the material being operated upon. With the strokes per minute settled, the rotary speed of the work is automatically fixed by the number of teeth required on the blank. It may happen that these factors being fixed, the form and slope of the spiral trace given by them are not quite what they should be. Correction, if not too much is required, may be sought by simply increasing or decreasing the length of stroke of the tool. An alteration in this factor leaves the pitch and number of teeth unaffected, and, within reason, does not greatly affect the cutting speed of the tool. The change, however, varies the amounts *GH* and *JK*, Fig. 1, by which the tool overshoots the work, and therefore varies the slope of the trace. In Messrs. Herbert's machine the overshoot, so far as the operator is concerned, is fixed at $^7/_{16}$ in., so that it is not within his power to vary the slope and form of the trace in the manner indicated.

If the mating wheel is an exactly similar pinion, the machine settings, once they have been satisfactorily found, will not be disturbed. If, however, a wheel is required which when mated with the pinion will give a speed-reduction ratio of R to 1, the rotary speed of the work spindle must be changed. In order to cut R times as many teeth as before, the ratio $2m/n$ must be made R times as great. The spindle speed n must therefore be reduced to $1/R$ of its former value. This alteration, if no change is made in the length of stroke or the speed of the tool, is incidentally exactly that required to generate precisely similar traces on the wheel and pinion. Since the widths of the wheel and pinion faces are the same, no change is called for in the length of stroke of the tool, even if such change were permissible. Moreover, since the speed of the tool and the peripheral speed of the work are the same as previously, the true cutting speed remains at its former value.

The article describes next the constructional details of the machine and gives a diagram illustrating the feed action. (*The Engineer*, vol. 140, no. 3631, July 31, 1925, pp. 112–114, 7 figs., *d*)

METALLURGY

Experiments in the Quench-Hardening of Steel Rails

A SUMMARY of some recent experiments in the quench-hardening or rail steel with the object of ascertaining a safe and inexpensive treatment for the hardening of the hardest-wearing portions of high-carbon switch and crossing rails. The experiments were conducted in a rather crude manner in the blacksmith shop of a permanent-way depot, with only such simple apparatus as blacksmith's fire, quenching trough, thermometer, horseshoe magnet, and a watch.

Until recently there was in England a standard treatment which applied to many thousands of rails manufactured under the old British Standard Specification to a maximum carbon content of 0.50 per cent, and it worked so long as the carbon did not exceed that limit. But as the carbon passed this and was ultimately increased to 0.65 per cent trouble began to be experienced from the process as it involved a drastic quenching, with the result that high-carbon rails in several instances cracked along the web under the nose and in the plane of junction between the hardened and unhardened areas of the rail. In other cases portions of the nose broke off under traffic. In the present case several methods were tried out. First was quenching out from above the critical temperature of the steel in water at approximately 100 deg. fahr., leaving the rail in the water until cold. This proved to be an absolute failure. Quenching the rail head only for a limited period, such as 90 sec., proved to be good in some cases but generally uncertain, with the further objection that serious distortion of each rail took place in the vertical plane due to the unequal cooling strains. (In this test only the head was actually in the water.) Potashing treatment was next tried, the rail being heated on the head to a blood-red heat only, after which the head was rubbed at this heat with yellow prussiate of potash, and was then quenched out in water at 100 deg. fahr. The results were very variable and the treated rails brittle—this in the face of the fact that potashing was successful when confined to rails whose carbon content did not greatly exceed 0.40 per cent.

Quenching in oil was tried, rape oil being used. This treatment again proved good for rails up to 0.5 per cent carbon, but was too drastic for higher carbons. Finally, quenching out with subsequent annealing was resorted to, and this proved to be quite satisfactory. In this there was complete freedom from brittleness and an average increase in hardness of 26.5 per cent. The wide range of treatment within which satisfactory results were obtained was also noteworthy, the variation in period of quenching from 60 to 90 sec. as also the variation in quenching temperature from 102 to 193 deg. fahr. appearing to have but little material effect in varying the test figures.

Quenching in boiling water without subsequent anneal was also investigated. It was found that the hardness grew steadily up to the 90-sec. quench when an increase of 25 per cent was obtained which is considered to be about the maximum obtainable with this treatment. Falling-weight tests proved that the steel so quenched had no trace of brittleness under impact, and subsequent fractures of the test pieces at the center showed a very satisfactory metallic structure very close in grain and homogeneous right down to the base of the head where the normal structure commenced.

After the conclusion of the experiments it came to the writer's knowledge that quenching, conducted on somewhat the same lines as those at which he arrived finally in these experiments, was being conducted on a commercial scale in a French rolling mill. The rails there manufactured, where the treatment is specified, are passed directly from the mill, by means of roller gear, at the finishing temperature, through a trough of boiling water, of sufficient length and at such a speed as will secure the quenching at the proper rate through the critical temperature range, after which they are passed on to cooling banks to cool in the ordinary way. Only the head of each rail is immersed. Satisfactory results have been obtained in this way, but it should be understood that the general maximum carbon limit worked to in the production of rails would not exceed 0.50 per cent, as compared with the 0.65 per cent limit to which the writer has worked in these experiments.

The limited-period boiling-water quench appears, therefore, to be a sound proposition for the treatment of the hardest-wearing parts of switch and crossing rails. It could hardly be simpler in operation; and, further, the method appears to offer a wide latitude, both in heats and in periods of quench, within which are obtained substantially increased hardness, and also adequate freedom from brittleness. Among the things discovered in the course of these tests it was found that lower-carbon bessemer acid rails gave harder test results than the higher-carbon rails of the same manufacture, although the differences in analysis were very slight, and also that the lower-carbon basic open-hearth rail gave comparatively low tensile-test results. The acid steel, both bessemer and open-hearth, proved harder under test than the basic for the same or a proportionate content. (Cecil J. Allen in *The Railway Engineer*, vol. 46, no. 546, July, 1925, pp. 221–226, *e*)

POWER ENGINEERING (See also Engineering Materials; Testing and Measurements: Measurement of Electrical Output of Large A.C. Turbo-Generators during Water-Rate Tests; Special Processes: Waste-Heat Recovery from Rotary Lime Kilns)

Generator Producing Steam from Water Spray

DESCRIPTION of the Becker steam generator which has no "water content" at all. In this case completely atomized water is injected into the tube system and immediately transformed into steam.

Fig. 2 shows a side elevation of the generator and furnace equipped for oil burning. The oil is atomized by blower *a* and an automatic pump forces the water through the water tube *c* into the pipes *d* where it assumes a rotary movement. It goes then into the coils located in the zone of the hot gases. The steam rises in the pipes *e* and is lead back to the vicinity of the fire zone to be dried and superheated. Then it passes through the exhaust pipe *f* into the steam reservoir *g*.

It is claimed that the space occupied by the Becker generator is only one-fortieth of the size of a tubular boiler of similar capacity.

The problem of boiler-scale formation is solved in so far as there is no crystalline scale. Ordinarily this occurs in the heating of water having more or less sluggish motion. Here, however, the water is transformed immediately into vapor while moving at high speed. Consequently, the scale is not crystalline, but is in the form of a fine dust which settles in the lowest tube sections of the generator. The lowest tube is extended beyond the rear wall of the generator and may be opened from time to time by removing two screws, so that the tube section may be taken out and cleaned. This is done by brushing and takes no more than two or three minutes.

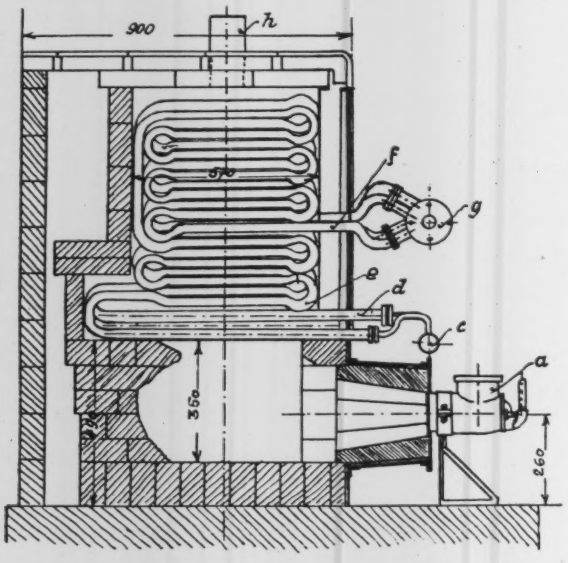

FIG. 2 BECKER FLASH BOILER

Its possibilities appear to be particularly interesting wherever steam is needed at regular times or, because of its rapid action, for peak loads. In new installations, or where plants are being remodeled and where exhaust steam is not utilized, high-pressure installations of 450 to 1500 lb. have been successfully tried out. (Franz Lang, Engineer, Vienna Institute for Scientific Auxiliary Work, in *Power Plant Engineering*, vol. 24, no. 17, Sept. 1, 1925, p. 893, 2 figs., *d*)

Smoke Prevention

THE Smoke Prevention Association held its 19th Annual Convention at Grand Rapids, Michigan, June 17–20, 1925. A day was devoted to railroad matters.

A. P. Mitchell, General Smoke Inspector, Pittsburgh District, Pa., said that it was only in August, 1924, that the smoke-prevention activities of the Pennsylvania Railroad in Pittsburgh were centralized under the general superintendent and the chief inspector appointed with authority extending over all divisions in the city. The organization of smoke prevention is described in some detail.

Two kinds of coal, bituminous or high volatile, and semi-bituminous or low volatile, are used in the district.

All locomotives in shifting, work-train, and interchange freight service use low-volatile coal exclusively. Each locomotive in passenger and freight service, working out of a Pittsburgh engine house, is supplied with 1000 lb. of low-volatile coal on the front of the tank to keep the fire in proper condition while lying at the enginehouse, to build the fire for the road, and to aid in preventing smoke while the engine is in what is called the "smoke district." This low-volatile coal is loaded after the fire has been cleaned or built, and the amount, 1000 lb., has been found to be sufficient, although there is now under consideration the possibility of extending its use to local trains serving the suburban towns near Pittsburgh. Low-volatile coal is used also in all power plants in the city whose boilers are fired by hand.

The saving due to the present method may be attributed to fuel, firing, and supervision. When the previous method was in use with high-volatile coal, the blowers on the locomotives were in constant use. While this tended to lessen the amount of smoke, the stacks were not as clean as they are with the low-volatile coal, and it was impossible to draw the line below Number 3 of the Ringlemann chart. In addition the continuous use of the blower would burn out the fire whether the engine was working or not, which meant that more coal had to be fired, making more smoke. More air also had to be admitted over the fire door, which of course had its effect on the firebox and flues. It has been found that 20 per cent more high-volatile coal is necessary than when low-volatile coal is used and a large percentage of this excess is burned because of the use of the blower, giving no beneficial result. With low-volatile coal, it is possible to keep the locomotive out in shifting service for 4 to 6 hr. longer than with the high-volatile coal, with fewer repairs to tubes and firebox. Getting up steam with fan or forced-air draft has been found to be more economical than with steam blower. The steam that is used to fire one locomotive, if converted into electric energy, is sufficient to fire three locomotives with fan draft. (*Railway Age*, vol. 79, no. 3, July 18, 1925, pp. 144–147, *gd*)

RAILROAD ENGINEERING (See also Metallurgy: Quench-Hardening of Steel Rails)

Sentinel-Cammell Steam Rail Cars

DESCRIPTION of two steam rail coaches recently placed in service on the London and North Eastern Railway. They are designed to carry 56 passengers. The power unit is self-contained and is easily detached for repairs. It consists of an engine, boiler, water tank, coal bunker, and the necessary gearing, and is fixed on an underframe having four wheels; both axles are chain driven from the engine. The engine develops 70 b.hp. and has two high-pressure double-acting cylinders, $6^3/_4$ in. diameter by 9 in. stroke, automatically lubricated. The vertical water-tube boiler supplies steam at 275 lb. per sq. in. which is superheated to 600 deg. fahr. The construction is such that by breaking two joints the firebox can be lowered and the inside of the boiler made accessible for examination and repair. (*The Railway Gazette*, vol. 43, no. 4, July 24, 1925, p. 149, 2 figs., *d*)

The Fireless Steaming System at Locomotive Terminals

WHILE the production of smoke at terminals not located near residential areas may apparently involve no direct penalty, it does affect the working conditions within the engine house and the elimination of smoke would be a distinct advantage in any location. This can be accomplished more easily than is generally appreciated by employing the recently developed fireless steaming method, and providing a firing-up shed adjacent to the engine house. This arrangement would result in segregating the fire cleaning, housing, and firing-up operations into three distinct positions at the terminal.

In the fireless steaming system a mixture of live steam and hot water is injected through the blow-off valve. When the boiler has been filled in this manner to a point where water shows in the coils, the water supply is closed and the flow of steam continued through the blow-off valve until the pressure in the boiler equalizes with the steam pressure in the supplying main or until a sufficient working pressure has been built up in the boiler.

To generate the desired steam pressure quickly it is necessary to supply steam at a pressure considerably higher than the pressure to be generated in the boiler as the rate of steam flow decreases as the pressures tend to equalize. It is also necessary to supply filling water at a temperature not less than 180 deg. fahr. so that the resulting mixture of water and steam entering the boiler will be well over 212 deg. Otherwise the steam would all be condensed instead of building up a working pressure. The operation of this method is therefore contingent upon a hot-water filling system and adequate live-steam generating capacity.

If these facilities are available, however, it is a comparatively simple matter to install the direct steaming system at any terminal by the addition of a live-steam main from the power plant to the blow-off and filling connections. The hot water and steam is combined in a special connection known as the direct steaming booster, and the resulting mixture is injected through the blow-off valve or a special connection in the base of the locomotive boiler so that the temperatures throughout the boiler while filling will

be uniform. If the steam and water were not combined or were injected into the upper part of the boiler, the steam would tend to segregate and accumulate in the top of the boiler before the water in the lower portion of the boiler had been raised to the temperature of the steam.

When the locomotive boiler is filled in this manner, the fire may either be lighted as soon as water shows in the glass or a steam pressure sufficient to move the locomotive outside the house may be generated without lighting the fire on the grates. If the fire is lighted immediately, the time required to fill and steam up locomotives can be reduced to about one-half hour. The time required to fill and steam up a locomotive to 100 lb. pressure in the manner described without lighting the fire is about one hour. Where the boiler is not emptied at the terminal, the locomotive can be held under steam for any length of time without a fire on the grates by maintaining a live-steam connection at the blow-off valve.

Attention is also called to the fuel economy resulting from the generation of steam in an efficient stationary boiler in place of steam generated in a locomotive boiler during the firing-up stage when its fuel efficiency seldom exceeds 40 or 50 per cent. Reduction in the use of blower steam is also mentioned, as with the fireless steaming method no house blower is required, and the locomotive blower is used less than when all the steam is generated by a fire on the grates. (L. G. Plant, National Boiler Washing Co., Chicago, Ill. Paper read before the Smoke Prevention Association, Grand Rapids, Mich., June 17–20, 1925, abstracted in *Railway Age*, vol. 79, no. 3, July 18, 1925, pp. 146–147, *dp*)

SPECIAL PROCESSES

Waste-Heat Recovery from Rotary Lime Kilns

THE advisability of installing waste-heat boilers on rotary lime kilns depends on the economy that may be derived from such an installation. It would appear that for every ton of lime burned there may be reclaimed by installing waste-heat boilers 55 hp. of energy, equivalent to 140 lb. of coal of 13,000 B.t.u. per lb. With coal costing, say, $5.00 per ton, this means of saving of 54 cents. Against this must be placed the expense of maintaining the boiler.

A number of factors have to be taken into consideration. The amount of heat which will be available from the kilns for the generation of steam depends on the market for the lime, which is frequently a very uncertain item. Another factor is the price of fuel at the particular plant and the regularity of the supply. After all, the business of the lime manufacturer is to make lime and if he can delegate the making of power to another he is relieved of a very considerable responsibility. In plants not laid out for waste-heat boilers there may be insufficient draft from the stack for both the kiln and an added boiler, which means the installation of an induced-draft fan. The fan may prove to be an advantage and it may not. When running it will assure a constant draft on the kiln, but if it fails to function the kiln must stop.

Should the stone used for making lime have a high content of surface moisture the heating of the boiler would not be as satisfactory as when the stone is dry. (Charles Longenecker in *Pit and Quarry*, vol. 10, no. 5, June 1, 1925, pp. 65–66, *p*)

TESTING AND MEASUREMENTS (See also Air Engineering: A Method of Determining the Dewpoints of Fuel-Air Mixtures)

Interference Tests on a Pitot Tube

BECAUSE of the universally felt uncertainty of air-speed measurements by the pitot tube, when the instrument is mounted close to some other body, a series of tests has been made in the No. 1 (5-ft. atmospheric) wind tunnel of the National Advisory Committee for Aeronautics with the object of determining the nature and magnitude of errors inherent to some of the most common installations.

Tests have shown that the maximum wear found does not apparently exceed 6.5 per cent in head corresponding to 3.4 per cent in velocity and that the scale effect is absent. (Elliott G. Reid. Report no. 199 of National Advisory Committee for Aeronautics, 1925, pp. 3–8, 10 figs., *e*)

Measurement of Electrical Output of Large A.C. Turbo-Generators during Water-Rate Tests

THE paper describes the methods used in measuring the electrical output in forty series of water-rate tests on machines ranging in rating from 10,000 to 45,000 kw. and also the application of photographic observations to such tests.

The water rate of large alternating-current turbo-generators is determined in place with the generator supplying power to the existing commercial load. It is therefore necessary that the electrical output be accurately measured under conditions where the load is practically always varying slightly and where it may be varying considerably.

A value of accuracy of the final result of the electrical output alone may be arrived at from a knowledge of the precision of the wattmeters and instrument transformers used. However, a more satisfactory accuracy figure is one which can be applied to the overall measurement of the water rate of the turbo-generator. This figure is the percentage average deviation from the mean, and is applied to the results obtained from three or more tests at any given load point. Enough tests can be run at any load point to bring the value of the percentage average deviation from the mean within any desired value. The percentage average deviation from the mean is given by a formula in the original article involving values of water rate obtained in the several tests.

The author describes the proper method of testing, instruments used, instrument connections, and processes for taking the observations. As regards the accuracy of final result, he states that a value within ± 0.25 per cent can be attained in water-rate tests where the measurements of electrical output are taken in the manner described and where equal care is given in making the steam-input measurements and corrections. This is confirmed by tables giving maximum difference between values of water rate obtained from tests at each load.

Among other things the author describes a method of photographic observations whereby Kidder and Hall of the Interborough Rapid Transit Co. of New York measured the electrical output of a turbo-generator by means of portable indicating wattmeters, and the observations were made with moving-picture cameras.

The photographic measurement was made by mounting an indicating wattmeter in a vertical position together with a moving-picture camera so focused that the scale of the instrument covered the entire width of the film. The wattmeter was provided with a special pear-shaped target pointer, and was properly balanced for operation with the shaft horizontal. Three such instruments and cameras were employed, one set for each phase. The shutter mechanisms of the cameras were operated in synchronism from a common motor-driven drive shaft. Number counters were so mounted on each instrument that the number appeared on the film, thus providing means for identifying all pictures. A watch was photographed on the film at the beginning and at the end of each test to obtain the rate of observation. Suitable illumination was produced from mercury-vapor tubes. The entire installation was mounted in the generator room near the machine under test.

The shutter-operating mechanism operated the shutters at one-second intervals for short-time duration. For long-time duration such as one-hour test periods, the fastest practicable rate was about one operation per seven seconds, which was the rate maintained during the water-rate tests.

Readings were obtained from the film after development by projecting the image on a screen through a projector and reading the value shown by the pointer. Readings were easily made to within 0.1 small scale division of the wattmeter scale, which is 0.5 watt. (Everett S. Lee, General Engineering Laboratory, General Electric Co., Schenectady, N. Y., in *Journal of the American Institute of Electrical Engineers*, vol. 44, no. 8, August, 1925, pp. 847–854, 3 figs., *ep*)

CLASSIFICATION OF ARTICLES

Articles appearing in the Survey are classified as *c* comparative; *d* descriptive; *e* experimental; *g* general; *h* historical; *m* mathematical; *p* practical; *s* statistical; *t* theoretical. Articles of especial merit are rated *A* by the reviewer. Opinions expressed are those of the reviewer, not of the Society.

Engineering and Industrial Standardization

Standardization in Japan

THE Japanese Engineering Standards Committee has been in existence since 1921. It is a governmental body functioning under the Bureau of Industry, Department of Agriculture and Commerce, Tokio, and is composed of seventy members, appointed or elected. Dr. Masawo Kamo, professor of mechanical engineering at the University of Tokio, contributed greatly toward its establishment, and his work is often likened to that of Prof. Comfort A. Adams, the first chairman of the American Engineering Standards Committee.

The introduction of standards into practice seems to be somewhat slow in Japan, due to the relatively undeveloped state of its industry, the greatest progress in that respect being chiefly among Japanese concerns having foreign connections. The disastrous earthquake of 1923 naturally had a detrimental effect upon the activities of the J.E.S.C., but notwithstanding the many difficulties experienced, as, for example, the loss of records, the last progress report lists 47 standardization projects under way, divided as follows: Civil Engineering 7, Mechanical Engineering 13, Electrical Engineering 7, Transportation 1, Shipbuilding 1, Ferrous Metals 8, Non-ferrous Metals 7, Chemical Industry 1, Miscellaneous 2.

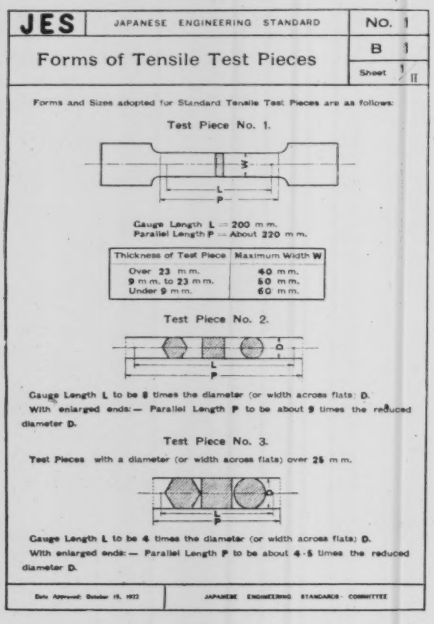
JES
JAPANESE ENGINEERING STANDARD
No. 1
Forms of Tensile Test Pieces
B 1
Sheet 1/2
Forms and Sizes adopted for Standard Tensile Test Pieces are as follows:
Test Piece No. 1.
Gauge Length L = 200 m.m.
Parallel Length P = About 220 m.m.
Thickness of Test Piece
Maximum Width W
Over 23 m.m.
40 m.m.
9 m.m. to 23 m.m.
50 m.m.
Under 9 m.m.
60 m.m.
Test Piece No. 2.
Gauge Length L to be 8 times the diameter (or width across flats) D.
With enlarged ends:— Parallel Length P to be about 9 times the reduced diameter D.
Test Piece No. 3.
Test Pieces with a diameter (or width across flats) over 25 m.m.
Gauge Length L to be 4 times the diameter (or width across flats) D.
With enlarged ends:— Parallel Length P to be about 4-5 times the reduced diameter D.
JAPANESE ENGINEERING STANDARDS COMMITTEE

FIG. 1

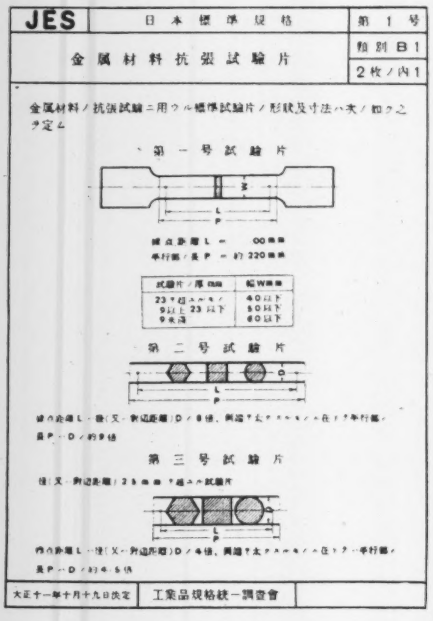

FIG. 2

Another dimensional single-sheet Japanese standard recently issued (No. 2, B 2) relates to the diameters of wire and thicknesses of sheet metals. Forty-two millimetric sizes are given, ranging from 0.10 to 12.00 mm.

In connection with valves, a draft proposal is under consideration, as is the case with four varieties of seamless steel tubes. Work is also well advanced on small three-phase induction motors, single-phase pole transformers, and electric lamp bases, while work will be undertaken on carbon brushes and insulating oils.

Draft standards for railway rails are under consideration. Steel for shipbuilding, standard bar stock, and standard sections are being standardized dimensionally, and tolerances on weights and dimensions of ordinary rolled steel are in the draft stage. Another important specification is that for foundry pig iron, which has been approved for publication.

Japanese Adopt Preferred Numbers

THE A.E.S.C. has just received from the Japanese Engineering Standards Committee two standard sheets covering preferred numbers, one entitled Preferred Numbers for Dimensions, and the other Preferred Geometrical Numbers. The first is a series progressing by increments which are systematic when regarded in the sense of the selection of digits, but quite unsystematic with reference to the rate of growth of the values from each number to the next. Thus the table begins by using numbers ending in 0.0, 0.2, 0.5, and 0.8, up to 3.5; then numbers ending in 0.0 and 0.5 up to 15, then every number to 26; every even number, but including also numbers ending in 5, up to 50; then numbers ending in 0, 2, 5, and 8 up to 100; in 0 and 5 up to 200; and finally numbers ending in 0 up to 500.

The application of this table as distinguished from the "geometrical numbers" is not evident, although it is supposed that the two tables have been developed following the precedent on the Continent where a table of standard diameters was first developed with the idea of reducing the number of sizes of gages, cores, tools, etc., used in any manufacturing plant to the minimum number without regard to the systematic nature of such progression, except from the point of view of numerical simplicity. Later, when the preferred-number series was proposed, it was felt by some that if the matter could be done over, the first or standard-diameter series would be eliminated because of the considerable duplication between the two series, and secondly, because it was realized that there would be a tendency in many cases for *all* the numbers of both series to come into use. It is understood that it has not been possible to annul the original standard-diameter series on account of the fact that many of those standard diameters have become crystallized in tool and gage sizes, so that considerable loss would be involved if they were unconditionally to be dispensed with.

The Japanese do not indicate the respective fields of these two standard sheets, and the A.E.S.C. is in correspondence with them to clear up the matter if possible.

The latest progress-of-work report of the Japanese covering the first half year of 1925 includes a total of 61 projects, of which 18 have been approved for publication. Among their projects of particular interest to American mechanical engineers are cast-iron water pipes, forms of tensile test pieces, and wire and sheet-metal gages (these latter two being published in this issue), steel tubes for boilers and for general purposes, files, twist drills, pulleys, spring steel, carbon brushes, steel sections, and plates and bars.

Standardization of Refrigerators Postponed

THE American Engineering Standards Committee announces that it has decided to postpone indefinitely the conference on specifications for household refrigerators, which has been under discussion the past two years. This decision is taken as the result of a request of the National Refrigerator Manufacturers' Association.

The subject originated with the A.E.S.C. in a formal request from the American Institute of Architects. There resulted a considerable amount of correspondence with interested organized groups—both governmental and industrial—among which it de-

veloped that there was considerable demand for recognized specifications. It appears that there is a wide variation in the ice efficiency of various refrigerators on the market, the less efficient consuming approximately twice as much ice for the same useful refrigeration as the most efficient types.

The purpose of the request of the American Institute of Architects was to bring about nationally recognized specifications as a guide to purchasers of refrigerators, which are now sold in a great majority of cases without definite knowledge on the part of distributor and consumer as to the ice efficiency and the degree of cooling of the food chamber with respect to the outside air temperature. The Federal Government has adopted specifications for its own use, but there are no generally recognized specifications.

The action in postponing the conference is in accordance with the announced policy of the American Engineering Standards Committee that it will not act as an initiating body, and that it undertakes work on a project only upon a formal request by a responsible body and then only after the Committee has assured itself that it is the consensus of industry that it is desirable that the work should go forward. Hence it was evident that the work on specifications for household refrigerators could not be prosecuted successfully under A.E.S.C. procedure so long as the National Refrigerator Manufacturers' Association is unwilling to enter a conference of all interested groups to discuss the advisability of undertaking the work, and in case of affirmative decision, to decide along what lines the work should proceed.

Correspondence

CONTRIBUTIONS to the Correspondence Department of Mechanical Engineering are solicited. Contributions particularly welcomed are discussions of papers published in this journal, brief articles of current interest to mechanical engineers, or comments from members of The American Society of Mechanical Engineers on activities or policies of the Society in Research and Standardization.

Indicating Weights of Castings on Blueprints

TO THE EDITOR:

As far as the independent jobbing foundry is concerned, indicating casting weights on drawings will convey that information to the foundry foreman only in a few cases, for the majority of the orders received by him are not accompanied with a blueprint.

Indicating the weights of castings by raised numerals on the pattern does not assure the foreman of correct weights. The writer has known of cases where a pattern has been rigged from one for a casting weighing 100 lb., to one that would give a casting weighing several times that amount. Then, too many of the patterns that enter a foundry making light castings are so constructed that there is not space enough available for so marking them.

The ideal way of informing the foundry foreman in regard to casting weights is to have all of the companies using castings indicate on their orders for castings the weight of each item. It is then an easy matter to have the weights appear on all standard forms used in putting the order through the foundry, and the shipping room, pattern room, office, and pouring room will all be informed as to the weights. Then, too, the weight of the casting can again be brought to the attention of the foreman by marking in a space reserved therefor on the pattern tag.

M. H. TIELKE.[1]

Cleveland, Ohio.

Factory Administration in India

TO THE EDITOR:

Factory administration in India differs in some respects with Western ideas, although the fundamentals and of course the ultimate aims are the same.

Coöperative Societies. Most natives of India, especially the natives of Bengal, are born in debt and die in debt, leaving a lifetime legacy of debt to their offspring. The Bengalee is, generally speaking, a peace-loving agriculturist, a fact which is taken advantage of by the more virile native of northwestern India—the Pathan—who lends him money at appalling usury and never allows the poor Bengalee or his family to get out of his grip.

With an attempt to get the Bengalee out of this dilemma cooperative societies are being introduced, worked in collaboration and under the factory executive, of which shares can be purchased cheaply, and to the value of the employee's credit in wages earned, but no credit is given beyond these earnings.

[1] The Crucible Steel Casting Co.

The workmen are encouraged to take shares in the society, paying for their shares by installments, and for which they get quite a reasonable rate of interest. They were, however, very suspicious at first about getting interest for apparently no effort, but latterly this is being appreciated. In factories where these societies are being introduced the workmen are getting more contented and can concentrate "orientally" on the job, instead of visualizing the Pathan usurer waiting outside the factories to squeeze a few rupees out of them.

Caste System. An executive must study the handling and blending of the various castes if he desires to be at all successful and get harmony and the loyalty of the Indian employees. In the workshops, castes of the same or similar standing must be blended together, and every effort made to avoid any interference, or even the suggestion of it, with their religious observances.

"Babus"—Indian Clerks. Give a Babu a routine or system to follow, and that routine or system is as the laws of the Medes and Persians; he will interpret and follow it to the letter, will learn it word by word; but ask him to deviate the slightest, or even to interpret an instruction routine or system different from what he has always been used to, he will look askance and shake his head: meanwhile one must give up attempting even with threats to get any deviation. However, in many respects this is a very good trait, which, if taken advantage of thoroughly, can be put to good use.

Labor. It is equally essential in India to organize a labor bureau in connection with factories, by which labor can be engaged, transferred from one department to another, or discharged. Where such a department does not exist, labor is engaged by the "mistri," who usually brings in his friends irrespective of their suitability, but those from whom he can get the most "baksheesh." This baksheesh being collected regularly, a proportion of the wages must be paid as long as the man is employed. If they protest or threaten to cease payment, the "mistri" arranges discharge or transfer to another department where the work is much harder. A labor bureau eliminates all this.

A complete record of service is maintained together with a photograph of the employee, as it has been found on occasions that a man has worked under one name, been discharged, and then applied later for work under another name, thinking the unsophisticated "white" would not know him again.

Mustering. Time clocks and similar western ideas are of very little use to the average illiterate oriental. A system of mustering is generally used, under a white or Eurasian supervisor, brass tokens or checks being hung on boards near the time office at the entrance gates, these being handed to men, who though not knowing their number, ultimately learn the position of the check on the board. On receipt of the check they proceed to their respective sections where they drop their checks into locked boxes provided for the purpose. Immediately after the starting whistle blows, and to prevent irregularities, these boxes are dispatched to the time office where the time records are compiled.

To prevent further irregularities and as an aid to costing, an

independent muster is made during working hours to ascertain the location of the men, and the jobs on which they were employed. If a man does not appear at the surprise muster, he loses pay for the whole day.

In addition to the ordinary workman there are other employees who consider themselves of superior class, such as chemists, clerks, and storekeepers. These are exempt from mustering because they have the intelligence to clock in.

Searching of Workmen. All workmen are subject to a superficial search, and on occasion to a thorough search, before leaving the factory. Most of the superior Indian staff are exempt, but must carry an exemption pass to be presented on passing out. Before the rigid inspection of exemption passes, the subtle oriental saw a loophole for irregularities, and used to loan his pass to a friend of the workman grade, which the latter frequently utilized as a means for petty pilfering. When this was discovered, any one not being in possession of his exemption pass forfeited the concession for three months, and was fined and compelled to pass through the search barrier with the workmen.

Absence without Leave. "Soldiering" is just as prevalent among orientals as westerners. Several cases have been discovered among the superior Indian staff where medical certificates had been obtained—these can be bought for 50 cents—to cover absence due to alleged illness or attendance at religious rites. Such cases are dealt with by immediate discharge.

Factory Storekeeping. The receipt and issue of stores is similar to the system prevailing in most western engineering concerns. Indian section storekeepers known as "godown" keepers are used throughout the factory. These are responsible for certain stocks, their issue, and the posting of stock ledgers or bin cards, but they are always quite willing to help a friend in another section without the usual paper transaction, so that on checking stock, which is done regularly, it is a frequent occurrence to find discrepancies. To obviate this, the godown keeper forfeits his annual increment, and probably his annual leave of 15 days, if, on checking the stock in any one section, there is a shortage over a predetermined value.

Inspection of Material. Unless a definite procedure is rigidly adhered to, and bulk is checked against samples, an attempt would be made to "palm off" the most shoddy material. Occasionally bulk material entirely different from the samples has been immediately rejected and the supplier, generally an Indian, duly notified to remove it at once. Unless informed that a charge for storage would be made if retained beyond a certain date, he would utilize the factory as storage ground for future customers, and bring such customers to view the material there.

Inventory. In addition to the usual methods of taking inventories a central system of inventory control has been introduced to allow of elasticity in use of stores held on inventory lists, and prevent excessive duplications by purchasing additional stock. Any section requiring an article or tool, which prior to the introduction of this system would be purchased, passes the requisition through a clearing office which keeps a record of all stores as a central inventory. If the required article or tool is found available in another section, a permanent or temporary transfer is effected.

Budgeting. The indirect-expenditure budget is computed on previous experience, but a review is made of all items and additions or reductions made to those considered necessary. Before a repair can be carried out in a manufacturing department, unless of extreme urgency, an estimate of labor and material is submitted by the repair or millwright section, and the comparison of estimated and actual expenditure posted on a form for the purpose, showing the allotment and the progressive position of the actual and enabling the department concerned to ascertain the fund position quickly.

Manufacturing Cost Control. For every manufacturing order issued, a progressive cost card shows estimated expenditure in labor and material, the actual weekly, the progressive total, and final actual cost. If the actual costs get dangerously near the estimated, the departments concerned are duly notified to take such action as will insure the estimates not being exceeded. In addition a summary is maintained showing the total value of all manufacturing orders issued, the weekly expenditure which is subtracted from the above giving a value of the work in hand in each department; from this the labor and time required to complete orders in hand are easily determined.

Apprentices. It is only on rare occasions a boy stays long enough to complete an apprenticeship course in India, as there are so many inducements in the native bazaar for boys who have had a year or two's training to earn a few cents more. Senior employees are encouraged to bring their boys to be trained. These boys are expected to be able at least to read and write in their own vernacular and to do simple arithmetic. Day classes teach the boys (*a*) their own vernacular, (*b*) English, (*c*) geometry, and (*d*) arithmetic. The full period of training is 5 years.

Bookkeeping. In certain factories where there is an Indian cashier, the executive, in addition to his numerous other duties, finds it necessary to check all cash entries and totals made by the cashier or his assistants and check the cash in hand periodically.

SAMUEL J. HOPPER.

Oak Park, Ill.

How Much Can Be Gained by Exhaust Turbines?

TO THE EDITOR:

The velocity of exhaust gases leaving the cylinder of an internal-combustion engine is persistently stated incorrectly, being based on a wrong calculation. On page 1107 of Marks' Handbook, second edition, 1924, it is stated: "During the establishment of pressure equilibrium, the velocity of the gases is very great, in the neighborhood of 2600 to 3000 ft. per sec." However, the gas velocity never exceeds 1800 ft. per sec. and is usually in the neighborhood of 1000 to 1200 ft. per sec.

Hütte, from which this statement evidently was taken, gives the formula by which this figure was arrived at:

"From the kinetic energy of the exhaust gases (transposed into English units)

$$778Q_a = \frac{1}{2}\frac{G_a}{32.2}w^2$$

the average velocity can be calculated, as

$$w = 223.7\sqrt{\frac{Q_a}{G_a}} \text{ ft. per sec.} \qquad [1]$$

Q_a, the heat content of the exhaust gases, can be determined the easiest way as the remainder from the heat balance or by transmitting the exhaust heat to cooling water." (G_a is the weight of the exhaust gases.)

The formula was originally stated by Güldner and is used by him in the same manner on page 39 in the third edition of his book on internal-combustion engines.[1]

From Formula [1] in the case of an oil engine, taking for Q_a 25 per cent of the fuel energy, $0.25 \times 18{,}000 = 4500$ B.t.u., and for G_a with 100 per cent surplus air, $1 + 2\,(13) = 27$ lb., we really do get $w = 2875$ ft. per sec.

This calculation, however, is based on the wrong assumption that the entire energy content of the exhaust gases is transformed into kinetic energy. If this were true, it would be theoretically an easy matter to construct a heat engine with a thermal efficiency greater than that of a Carnot cycle. We utilize 40 per cent of the fuel energy in the cylinder, leaving a remainder of 60 per cent which should go out with the exhaust. If this could all be transformed into kinetic energy and drive a turbine, the efficiency of which in the ideal case is 100 per cent, the efficiency of the combination would be $40 + 60 = 100$ per cent, which is greater than the Carnot efficiency.

As a matter of fact, only a fraction of the exhaust energy can ever be converted into kinetic energy. Fig. 1 shows an ideal Otto cycle with a work area 01230. At the end of the expansion the state of the gas is represented by point 3. By converting the exhaust energy into kinetic energy additional work can be gained. The same end could be attained by letting the gas expand further in the cylinder. The work area gained by prolonged expansion is 03A0 in Fig. 1. It can be proven that the work gained in this manner is equal to the work converted into kinetic energy, when

[1] Güldner, Entwerfen und Berechnen der Verbrennungskraftmaschinen, 1914.

the exhaust gases blow out into the atmosphere, disregarding the friction losses.[1]

But the area 03A0 does not represent the total energy contained by the exhaust gases. The entropy diagram, Fig. 2, offers a convenient means for estimating the loss, on which 01230 represents the ideal Otto cycle. The area $S_1 12 S_2 S_1$ represents the introduced heat energy (fuel energy) and area $S_1 03 S_2 S_1$ the exhaust heat energy. How much of the latter can be converted into mechanical work?

According to the second law of thermodynamics, at each conversion of heat into mechanical work the amount of unavailable energy is the lowest absolute temperature of the cycle times the increase in entropy. Accordingly the area $S_1 0 B S_2 S_1$ in Fig. 2 is an unavoidable loss, both theoretically and practically, and only the additional area 03B0 could be utilized. However, this would involve an adiabatic expansion to point B. The corresponding pressure p is smaller than p_0, hence subatmospheric. Such an expansion could only be carried out with a "condenser." For practical

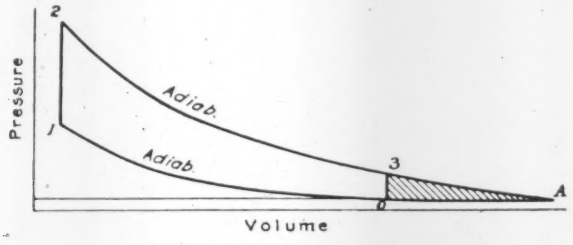

FIG. 1 P-V DIAGRAM OF OTTO CYCLE

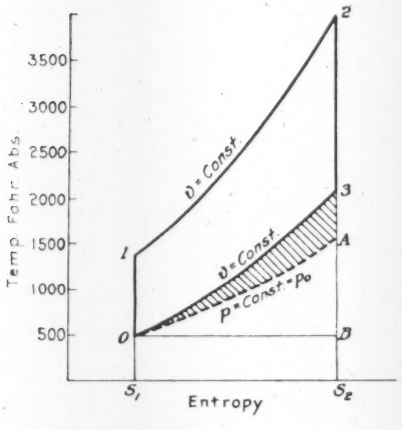

FIG. 2 T-S DIAGRAM OF OTTO CYCLE

purposes the expansion can only be carried to point A which corresponds to the atmospheric pressure p_0. Consequently the area 0AB0 cannot be utilized. The work that may be gained is only the shaded area 03A0. This corresponds exactly to the area 03A0 in the P-V diagram, hence the areas are equal. Fig. 2 is drawn approximately to scale and the proportion of the available energy to the total exhaust energy can be estimated. It can also be calculated as follows.

The heat change along the line 3-0 is

$$\text{Area } S_1 03 S_2 S_1 = \int_0^3 TdS = \frac{A}{k-1} v_0(p_3 - p_0)$$

and along A-0

$$\text{Area } S_1 0 A S_2 S_1 = \int_0^A TdS = \frac{Ak}{k-1} p_0(v_A - v_0)$$

The available portion of the exhaust energy compared with the total:

$$\eta = \frac{\text{Area } S_1 03 S_2 S_1 - \text{Area } S_1 0 A S_2 S_1}{\text{Area } S_1 03 S_2 S_1}$$

$$= \frac{\frac{A}{k-1} v_0(p_3 - p_0) - \frac{Ak}{k-1} p_0(v_A - v_0)}{\frac{A}{k-1} v_0(p_3 - p_0)}$$

$$= \frac{v_0(p_3 - p_0) - kp_0(v_A - v_0)}{v_0(p_3 - p_3)}$$

From the adiabatic change

$$p_0 v_A^k = p_3 v_2^k$$

where p_0 is the atmospheric pressure, 14.7 lb., and p_3 is the pressure in the cylinder at the opening of the exhaust valve.

$$\text{Hence } \eta = 1 - \frac{kp_0}{v_0(p_3 - p_0)} \left[\left(\frac{p_3}{p_0} \right)^{1/k} v_0 - v_0 \right]$$

$$= 1 - \frac{kp_0}{p_3 - p_0} \left[\left(\frac{p_3}{p_0} \right)^{1/k} - 1 \right]$$

With $k = 1.4$ for various exhaust pressures we obtain

p_3 =	30	40	50	60	70 lb. per sq. in. abs.
η =	0.11	0.15	0.18	0.21	0.24

The recoverable part of the exhaust energy is between 10 and 20 per cent with ordinary exhaust pressures. That is all that can theoretically be converted into mechanical work and utilized in an exhaust turbine or any kind of energy transforming medium. The rest of the heat energy remains in the gas and is dissipated finally in the atmosphere.

The velocity of the exhaust gases is considerably smaller than obtained by Güldner. Assuming that all of the convertible energy is transformed into velocity, we get (see Marks' Handbook, p. 366)

$$w = 223.7 \sqrt{(i_1 - i_0)} \text{ ft. per sec.}$$

where $i_1 - i_0$ is the convertible part of the energy of 1 pound of the exhaust gases. In the case of an oil engine each pound of the exhaust gases contains about $^1/_{27}$ pound of burnt fuel which represents 18000/27 = 667 B.t.u. fuel energy. Assuming that 25 per cent of this goes out with the exhaust, $i_1 = 0.25 \times 667 = 167$ B.t.u. The convertible part of the exhaust, however, is, as has been determined, about 20 per cent of that amount, so

$$i_1 - i_0 = 0.2 \times 167 = 33.4 \text{ B.t.u.}$$

Consequently

$$w = 223.7 \sqrt{33.4} = 1300 \text{ ft. per sec.}$$

With gasoline engines and blast-furnace-gas engines the corresponding figures will be 1700 and 1450 ft. per sec., respectively.

These are the theoretically possible exhaust-gas velocities. However, in an actual engine not even these velocities will be realized. The reason is that the velocities derived are greater than sound velocities and can be attained only in divergent nozzles. The exhaust valves or exhaust parts of an internal-combustion engine cannot be considered as divergent nozzles and the issuing gases will not exceed the corresponding sound velocities. The formula gives for the maximum velocity through a plain nozzle

$$w_{\max.} = \phi \sqrt{\frac{2gk}{k+1} p_1 v_1}$$

or with

$$p_1 v_1 = RT_1$$

$$w_{\max.} = \phi \sqrt{\frac{2gk}{k+1} RT_1}$$

and with $k = 1.33$, $R = 54$, and $\phi = 0.8$, $w_{\max.} = 35.7 \sqrt{T_1}$.

The exhaust velocity depends on the absolute gas temperature and is for

t =	800	900	1000	1100	1200 deg. fahr.
$w_{\max.}$ =	1196	1316	1364	1410	1454 ft. per sec.

In calculations with exhaust-gas velocities these figures should be used instead of Güldner's figures which are several times higher, and it also should be remembered that in exhaust-gas turbines not more than 10 to 20 per cent of the exhaust energy can be recovered.

P. H. SCHWEITZER.[1]

State College, Pa.

[1] It is to be noted that the proposed expansion of gases takes place from a source under varying pressure, not constant pressure, so as to preserve for the gas engine practically the original cycle 01230.—EDITOR.

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MECHANICAL ENGINEERING

A Monthly Journal Containing a Review of Progress and Attainments in Mechanical Engineering and Related Fields, The Engineering Index (of current engineering literature), together with a Summary of the Activities, Papers and Proceedings of

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The Beginning of American Air Transport

FOR SEVERAL years American aviation in the matter of planes, engines, and general technical development has been abreast of European practice, or even ahead of it; but it has been customary to bewail the fact that in spite of every geographical and economic advantage necessary to success in air transport, we have lagged far behind Europe in the commercial application of the airplane. Yet in Europe distances are comparatively short; air traffic is hampered by the many international boundaries, customs barriers, and varying regulations; and the value of saving time is not regarded as highly as in the United States. The reason for European superiority was attributed quite correctly to the fact that military considerations had led to every possible form of Government support, and in particular to the granting of generous and well-planned subsidies. Competent observers are agreed that in spite of the absence of subsidies, American air transport is now coming into its own, and on a more solid basis than is the case abroad. The awakening of interest is due to a number of very definite reasons.

If the annual national exodus eastward has served any purpose at all, it is that of bringing home to the American public that a great air-traffic development was indeed in process in conservative Europe. American tourists, particularly in the summer months, constituted the majority of the air travelers between London and Paris, London and Amsterdam, and on other routes. At home the Air Mail had functioned for several years with success, but without an overwhelming advantage in time saving. Only when, for the first time in the history of aviation, night flying was achieved in the country which had seen the birth of the airplane was this overwhelming advantage secured. When a letter posted in New York in the evening could reach Chicago the next morning, or San Francisco the morning of the second day, bankers and business men generally awoke to the tremendous possibilities of the Air Mail. The Air Mail was, however, a purely governmental enterprise. American capital was still apt to regard flying as a hazardous undertaking financially speaking, although certainly worth watching for its future possibilities. The few and scattered commercial air transport companies had met with very dubious results. It was the passing of the Kelly Bill which provided the final argument in favor of air transport. By this bill the Postmaster General was authorized to let contracts for the carrying of air mail to private enterprise. Passenger carrying still remained a highly doubtful undertaking, but the possibility of actually obtaining Government contracts for mail, and the highly favorable attitude of the American Railway Express, coupled with the success of the night airway across the continent, broke down the resistance of American financiers.

A great number of such air transportation companies are now in process of formation. Henry Ford is operating on a daily schedule from his splendid airport at Dearborn, Michigan, to Cleveland and Chicago. The National Air Transport Co. with Col. Paul Henderson as its operating head has sound financial support and is definitely planning a New York–Chicago air line which will begin operations some time early next year. Other equally serious interests are studying routes across the United States, coast lines to Florida, and even to Central and South America.

It is quite evident that the problem of securing financial support has been solved. The securing of traffic in the form of mail and express matter will be a comparatively simple matter provided adequate reliability as well as speed can be secured. The future of air transport reverts, therefore, to the mechanical and industrial engineers. Can the mechanical engineers (and aeronautical engineers may be included in this category) provide reliable planes and motors? Can they solve the problem of flying through fog and cloud when the lighted airway fails to guide the pilot? Can the industrial engineers so organize the airports, airways, and ground personnel that maximum reliability is not coupled with prohibitive costs? We believe that they can, that they have already eliminated the difficulties in the way to a large extent. American business coupled with American engineering cannot fail to succeed.

There is only one serious danger in this important and epochal development, and that is the lack of legislation and of the kind of Government support that has been so generously extended to every other form of American transportation. Air transport companies do not demand direct subsidies which are contrary to national feeling, but they do demand congressional action which shall define their liabilities and privileges, particularly in such matter of right of way, damage to third parties, and other important points. Further, they wish to secure Federal aid for the construction of aerial lighthouses and the charting of airways, and state and municipal aid in the construction of airports which private interests may use for financial considerations just as they use city harbors and docks. Careful study of governmental and municipal relationship to shipping will prove beyond cavil that similar aid for air transport is legitimate and certainly necessary.

If Congress next December turns its attention to these matters, the last serious obstacle to the growth of air transport in the United States will be removed.

ALEXANDER KLEMIN.[1]

Frontiers but no Boundaries

THE continued success of the coöperative research activities of the Bureau of Mines in this country and the Safety in Mines Research Board of Great Britain demonstrates the fact that while science may have its frontiers, it knows no international boundaries. Under the working agreement of these two research organizations, it is possible to save duplication in investigations which require both time and money; to increase the efficiency and interest of the research workers of the two bureaus by the opportunities afforded them of exchanging ideas; to obtain a greater variety of testing possibilities as is done by the exchange of coals and the corroboration of the deductions of one staff by another working under different conditions; and to increase the practicable applications of safety devices and the publicity of the hazards and safeguards of the mining industry.

The British investigations of the explosibility of coal-dust mixtures have been carried on in the testing gallery at Eskmeals in a surface chamber having smooth sides and a circular cross-section, while those by the Bureau of Mines have been made in the experimental service at Bruceton, Pa., which has rough sides of a rectangular cross-section and is of full-scale dimensions. Dr. R. V. Wheeler, of the Department of Mines of Great Britain, on

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a visit to this country during the spring, was able to correlate his coal-dust explosion experiments with those of George S. Rice, Chief Engineer, U. S. Bureau of Mines. It was found that coal shipped from Great Britain behaved under test almost the same as Pittsburgh coal as the composition is almost the same. The test showed that the difference in English and American test results is due to the methods of testing.

Under the coöperative agreement, the actual mine tests are to be conducted at Bruceton, while the gallery and laboratory tests will be continued by Dr. Wheeler in Great Britain and by the Bureau of Mines in this country. Confirmation on a practical basis of laboratory results will thus be obtained at the experimental mine.

During the past year Mr. H. P. Greenwald, assistant physicist of the Bureau of Mines, was detailed to Great Britain to work under Dr. Wheeler. While abroad Mr. Greenwald made investigations of some of the rock-dusting methods emphasized in British Mines. Mr. A. C. Feldner, supervising chemist of the Bureau, also made an extended visit in Great Britain, partly with the view to correlating methods of testing coals and laboratory work in connection with coal, coal dust, and gas.

Engineers will approve of the highly practical attitude which these two research bodies have assumed, and will encourage its continuation and extension.

Supersaturation

SUPERSATURATION is one of the phenomena of retardation of change of state. We are apt to think of the freezing point (temperature), the boiling point (pressure and temperature), and the saturation line as sharp and clear boundaries between solid and liquid, liquid and vapor, vapor and gas. Really, in certain conditions they may become slightly blurred or may be in some degree overpassed.

For example, if a body of water is kept very quiet and heat is slowly allowed to escape by conduction and radiation (without convection) it may be supercooled as much as 5 to 15 deg. fahr. below the freezing temperature. In like manner and amount a body of liquid may be superheated above the vaporization temperature for the existing pressure. In either case a slight mechanical disturbance will break up the abnormal state and the substance will come to physical equilibrium, with the sudden formation of a little ice or of a little steam.

For present purposes the saturation line is the lower limit of the superheated or gaseous state of steam. Under constant pressure the abstraction of heat makes the temperature fall until the saturation point is reached; then any further escape of heat causes the steam to condense, at constant temperature. In other words, steam of a given pressure cannot exist below the saturation or vaporization temperature for that pressure—at least, not in a state of equilibrium.

Supersaturation manifests itself when the temperature of steam falls below that of saturation, either without condensation beginning or with condensation lagging behind the full amount for equilibrium. It occurs in rapid adiabatic expansion and pressure drop into the region of wet steam. Equilibrium requires that as the operation proceeds the steam shall condense in increasing proportion, in order to yield the heat equivalent of the external work done. If condensation lags this heat must come from a cooling of the whole body of steam.

There is clear evidence that such cooling does occur, ranging to as much as 30 or 40 deg. fahr. An obvious way to detect and measure it would be by simultaneous readings from a search tube and a delicate thermocouple in the steam current. Unfortunately the temperature measurement fails because of surface effects, an adherent film of liquid and impact action being offered as explanations. Since direct evidence is not available it is necessary to resort to inference.

The simplest indication of supersaturation is found in the fact that actual discharge of saturated steam through an orifice or nozzle is greater than the theoretical amount calculated on the assumption of expansion in thermal equilibrium. Explanation can most easily be made by means of a few computed values.

Starting at a dry-saturation point, the ideal adiabatic is closely represented by using $n = 1.135$ in the equation $pv^n = C$. In the region of superheat the value of n is from 1.3 to 1.33. Suppose that, with supercooling, the exponent 1.3 continues to prevail below the saturation point. Calculated down to the throat pressure, or with $p_2 = 0.58\ p_1$, comparative values with $n = 1.3$, as against unity for $n = 1.135$ in each case, are as follows:

Specific volume	0.941 or −5.9 per cent
Work into kinetic energy	0.971 or −2.9 per cent
Velocity	0.985 or −1.5 per cent
Area of unit jet	0.955 or −4.5 per cent
Discharge, given area	1.047 or +4.7 per cent

Recorded experiments show an excess ranging from 1 to 4 per cent in discharge.

The really interesting question, which can be no more than suggested here, is, How does condensation catch up with heat demand? Does it hold back until a certain state of limiting tension is reached and then take place with extreme rapidity, or does it begin sooner and occur more slowly, with the effect of a continued effort to catch up? All things considered, the second idea appears the more probable; there is belief and some positive evidence that an appreciable degree of supercooling continues clear into the exhaust outlet of a steam turbine, which, however, may not involve a very large ratio of expansion below saturation. Remember that "rapid" and "slow" are comparative terms, since the whole time of passage of steam through a big turbine is only from 0.01 to 0.02 sec.

R. C. H. HECK.[1]

Intelligent Wood Conservation

SOMETIMES a public evil is so vast, so obvious, and so menacing that public attention to it, though beginning with the merest apathy, rapidly grows into something like fierceness. There arises an insistent demand for reform, and newspapers, special writers, philanthropists, and women's clubs vie with each other in denunciation, and in demand for reformatory action by the state or national government.

Unfortunately, it is one thing to be indignant over a bad state of affairs and quite another thing to know what can really and practically be done to reform it without involving even greater harm. Unfortunately, too, the reasoner who demands that an indignant public stop to consider before rushing into precipitate action, is usually regarded as a reactionary and a tool of predatory interests. All of which is absurd, but true.

The lumber industry of this country is at present under the microscope of public attention in its treatment of the nation's forest resources. The American Society of Mechanical Engineers has within recent years begun to take in that industry the interest to which it is entitled by reason of the increasing complexity of the engineering problems involved in the manufacture of lumber. The Wood Industries Division of the Society is studying every phase of the questions involved in the production of lumber, the conservation of lumber resources, and especially the elimination of waste. Only by such preliminary study can facts be uncovered in sufficient quantity and quality to satisfy engineering standards and insure results scientifically and mechanically worth while.

The engineer cannot rush blindly into denunciation of the lumber industry merely because he can discern the forthcoming rapid exhaustion of the nation's standing timber. He, at least, has a professional conscience, and that conscience demands that he first find out what the industry has to say for itself. It happens that the lumber industry has a spokesman in Mr. John Blodgett, of Grand Rapids, past-president of the National Association of Lumber Manufacturers and one of the ablest lumber men in the country. Mr. Blodgett is a conservationist and a reformer. He discerns evils and he frankly uncovers them; but he is a practical man and he talks common sense. At the annual meeting of the Society in December, 1923, this able industrial thinker read a paper at the general session which clearly set forth the appalling situation of the nation with respect to lumber resources, but which at the same time showed that conservation is a

[1] Professor of Mechanical Engineering, Rutgers College, New Brunswick, N. J. Mem. A.S.M.E.

matter of public intelligence, not public anger; and especially a matter of obtaining the practical assistance of the engineering profession in establishing standards and eliminating waste in the use of wood.

Mr. Blodgett is now once more putting us in his debt. Before the annual meeting of the Chamber of Commerce of the United States, held in Washington last May, he again attacked the subject. He showed conclusively that, instant as is the need for conservation of the remaining supply of timber, the question of practically securing this is extremely complicated. Selling conditions are fiercely competitive, while the industry generally is overwhelmed with an ever increasing burden of local taxation, almost compelling the manufacturers of lumber to produce at utmost speed, to throw their product upon the market at cut-throat prices, and thus to waste the remaining stands of timber without benefit either to themselves or to the consumers. It is evident that to preach conservation in these circumstances is futile—unless and until the public conscience is awake to the economic conditions which are throttling the lumber industry. When the people understand the economic facts, and not until then, the politicians will be compelled to revise their present unscientific and harmful taxation methods.

Meanwhile, however, Mr. Blodgett does not fail to show that the consumers have an equal responsibility with the manufacturers and the governments. For practical purposes the consumers of lumber, those with whom the manufacturer deals directly, are the architect and the engineer, who dictate what shall be bought by the home maker, for house, furniture, and the innumerable wooden articles which are essential to civilized life. It is they who must learn to standardize, to accept wood cut in odd dimensions, and thus to shoulder their part of the common responsibility. The engineer has before him the job of coöperating with the lumberman in putting to use some or all of the from twelve to eighteen per cent additional recovery from the average log which competent judges estimate could be made if scientific dimension standards were erected. Standard grades, grade names, and dimensions are among the crying needs of the situation. Engineers alone can establish and maintain such standards. The Wood Industries Division of this Society is fully awake to these facts and earne tly asks the coöperation of every member in this its so vastly important work.

Mr. Blodgett has done the profession a service in showing how in one important question of public policy a vast amount of public education in the facts is urgently needed; while engineering methods must, as they can, be brought to bear upon the practical questions of manufacture and consumption. If government, engineer, and manufacturer will get together, the problem will be solved and American timber resources will not be wholly exhausted within a quarter of a century.

WILLIAM BRAID WHITE.[1]

Tidal Power

AT A SPECIAL election on September 14, 1925, the voters of the State of Maine endorsed the project proposed by Dexter P. Cooper for generating power from the tides of the Bay of Fundy. Mr. Cooper's scheme consists of separating two sections of Passamaquoddy Bay, a tributary of the Bay of Fundy, by a wall on which the power house is erected. Each subdivision of this bay is accessible to the Bay of Fundy, but is to be shut off by suitable dams with locks and gates so that one part may be maintained at a high level by opening the gates to the Bay of Fundy only at high tide, and the other part kept at a low level by opening its gates only when the Bay of Fundy is at low tide. The turbines will be operated by the flow from the high-level to the low-level section of Passamaquoddy Bay. Mr. Cooper estimates that from 500,000 to 700,000 hp. may be developed and that the cost of the project will be approximately $75,000,000.

Mr. Cooper is thus well launched on a fascinating engineering adventure that involves an attack on a problem which has puzzled several generations of engineers. It is a simple matter to calculate the enormous energy that might be generated if the work of the earth's satellite on the Seven Seas could be converted into electric current, but thus far the results of all endeavors to effect such a conversion have been discouraging. Mr. Cooper, however, has the best natural site on this continent for using tidal power, and engineers will look on with encouraging interest as he proceeds in his struggle to generate and market large blocks of power, confronted as he probably will be, with high construction, installation, and transmission costs, and with competition from electricity generated at constantly reducing costs by steam prime movers.

In 1920 a tidal-power development was proposed at Hopewell, New Brunswick, on the Shepody Bay. Two main dams were proposed. The western dam was to be provided with automatic flap gates open upstream which would allow the high-level basin to be filled at each high tide, and at the eastern dam there were to be similar gates opening downstream to allow the low-level basin to empty on every ebb tide. At Hopewell about 15 per cent of the time the tides exceed 42 ft. range.

Another project to which a great deal of careful study has been given in England is a scheme for utilizing the tides in the Severn estuary. This was investigated by Sir Philip Dawson and Prof. A. H. Gibson, acting as a sub-committee of the Water Power Resources Committee of the British Board of Trade. It was found that under certain conditions the scheme could be carried out and might pay for itself, but that it involved enormous difficulties in various ways. While a considerable amount of money was spent in the investigation of the problem, nothing has been actually done since.

In the Severn investigation it was decided that at least part of the load could be sold to the cities within reasonable districts from the estuary and that this would constitute a high-paying load which would permit selling commercial current at attractive prices. Investigation of several chemical industries using large quantities of electrical energy for nitrogen fixation, etc. has shown that they are profitable only provided the cost of current is low, such figures as $7 to $10 per horsepower per year having been mentioned in the past. Were the Bay of Fundy within easy transmission distance from big cities like Boston or Providence, the question of market for power would be comparatively simple.

The Shenandoah

THE *Shenandoah* tragedy, deeply regrettable as it is, is a proof of the courage of her crew, men whose remote ancestors went down to the sea in ships that were at the mercy of the waves as the *Shenandoah* was before the Ohio gale.

It is inevitable that the *Shenandoah* disaster must contribute to progress in aerial navigation. The official investigation should reveal some facts that may be built upon in dirigible construction, which presents some problems that have no parallel in other engineering designs.

After the collapse of the British dirigible *ZR-2* over the Humber in August, 1921, in which disaster sixty-two British and American engineers and officials lost their lives, MECHANICAL ENGINEERING commented editorially as follows:[1]

> There is good reason to believe that the fundamental cause of the airship disaster lies also in lack of knowledge of vital elements underlying the design of large airships. It is, at times, difficult to realize how slight our knowledge of airships engineering really is. We are dealing with structures 600 to 700 ft. long, weighing in the air next to nothing. At both ends of these immensely long structures we have operable planes (rudders and elevators) of considerable size, presenting resistance to the air equal to a pressure estimable in tons, which, with a leverage of some 300 ft., must impose tremendous stresses amidships. What these stresses are we do not know, nor have we either experimental or mathematical bases for computation. We do not even know to what extent the theoretically rigid dirigible is capable of flexure.
>
> In a bridge, an ore boat, an automobile, generous factors of safety are used wherever there is doubt as to the stresses to which a member is likely to be submitted, because there is no vital gain outside of the cost consideration, which should be secondary in using excessively light members. But this is not so in an airship. If the latter is designed to fly across the Atlantic it must carry a certain weight of gasoline, oil, and useful load, and every pound of these supplies reduces by a pound the weight of the metal that can be put into the structure, and hence the factor of safety, with the result that members one-sixteenth to one-eighth of an inch in thickness are by no

[1] Chairman Wood Industries Division, A.S.M.E.

[1] MECHANICAL ENGINEERING, vol. 43, no. 10, October, 1921, p. 693.

means uncommon in dirigible construction; and members of such slender dimensions in duralumin, under the tremendous stresses they are called upon to withstand, no longer possess a factor of safety but rather a factor of daring.

In 1922 the *Roma*, which the United States bought from Italy, fell out of control, struck electric wires, and exploded, killing thirty-eight people, and the *C-2* disaster occurred at San Antonio, Texas, with a loss of several lives. In 1923 the *TC-1* was destroyed by a storm under conditions which indicated inherent weakness of the vessel's structure.

In the *Shenandoah* unusual pains were taken to make the dirigible strong. Pressures all over the surface under ordinary conditions were determined by special apparatus developed at Langley Field. Because of the suspicion that in the design of the *ZR-2* in England dynamic stresses had not been given sufficient consideration, special committees on stresses were formed under the auspices of the National Advisory Committee for Aeronautics and the assistance of prominent bridge and structural engineers was secured. There is therefore every reason to believe that the *Shenandoah* embodied the utmost in the way of design and safety known to engineers today, and yet she failed under the unusual stress of storm conditions.

It may be claimed, of course, that more careful weather observations should have been made and that the airship should not have been sent over the Ohio Valley where violent storms are known to be prevalent during the period between the middle of August and the latter part of September. This, however, is beyond the point, for an airship intended for use over long distances must sooner or later run into a severe storm, and no amount of weather observation will protect it from such an eventuality. A better-organized meteorological service, such as is found on some air routes in Europe, making frequent reports to the airship and the designing of the ship for greater speed to permit escape from forecast areas of disturbance will greatly reduce the recurrence of the *Shenandoah* disaster.

Because dirigibles are believed to have a certain military value, further experimentation with this type of vessel may be expected. The airship as built today, however, is a fair-weather-machine and much too delicate in construction to weather violent atmospheric disturbances.

The Ohio "twister," an ugly customer indeed, has been known to tip up a house and carry it 100 ft. away, or to lift a roof and carry it a mile, and yet leave houses to the right and left of the one wrecked perfectly unharmed. This means that there is an air disturbance traveling at the rate of some 125 m.p.h. but only a few hundred feet wide. Any reader can estimate for himself what would happen to a structure 600 to 700 ft. long, weighing in the air next to nothing, with a volume measured in hundreds of thousands of cubic feet, built of slender stampings, and with every possible pound of weight eliminated, when hit, say, amidships by such a cyclonic wind. In the wreck of the *Shenandoah* it is to be noted that while the control cabin was carried away, the ends of the ship were apparently unharmed.

Dirigibles without doubt have their place in the economy of things, but much further experimental work must be done in the direction of their development, for it should be clearly understood, as was stated four years ago in the editorial in MECHANICAL ENGINEERING cited above, that in a dirigible as built today under storm conditions there is no factor of safety but only a factor of daring.

Recognizing the Exceptional Student

AN INTERESTING innovation in engineering education is to be inaugurated this fall by the Electrical Engineering Department of the Massachusetts Institute of Technology in the formation of an Honors Group of selected men of the class of 1927 for the purpose of encouraging a scholarly attitude toward scientific and economic studies and enlarging the opportunities for breadth of knowledge and the exercise of vision, originality, initiative, and resourcefulness. The method by which this is to be accomplished, and the principal features of the plan, are outlined in letter by Prof. Dugald C. Jackson to the dozen men chosen by the faculty as being most likely to profit by the new arrangements. Excerpts from the letter follow.

The plan, very briefly stated, provides to the students in the group greater independence of study within the scope and ideals of the curriculum than is characteristic of our usual practice, and therefore includes the following features:

1 The students in the group will be privileged to attend the class exercises of the regular subject or not, as they individually please; but will be expected to successfully pass the usual term examinations, which should be readily accomplished as a result of the special reading proposed.

2 Of home problems assigned to the classes, the students will be privileged to work out such number as they please, but each individual will be encouraged to spend time over the details of only those that illustrate principles new to him, thus avoiding loss of time by any unnecessary repetition. All will be expected to do the work in the regular weekly problem section.

3 The laboratory work of each term which now usually consists of a series of independent assignments, will consist of a general assignment relating to the principles of construction and the characteristics of the circuits, instruments and machinery treated during the term, with the time and method of work largely determined by the interest of each individual student, who will carry on under the advice and direction of the Conference Adviser referred to below. Students will be encouraged to carry on this work as far as practicable as individual investigations of the principles and applications under consideration, directed along paths of their own interests in the subject. The usual laboratory reports will be omitted, and the report for each student will be his notebook containing an outline of his plan for the term, the record of his various investigations and measurements, his comments, and a brief summing up toward the end of the term of his accomplishments and progress during the term.

4 Freedom from existing restrictions of scheduled class hours and laboratory hours will afford greater opportunity for more reading and study related to the subjects under consideration. In order that the students' progress may be orderly and any difficulties encountered may be courageously faced and overcome a conference of an hour and a half each week will be held between the group and a member of the faculty learned in the subjects for the term, in which conference the progress and the difficulties will be mutually discussed. Substitution of subjects in the curriculum will also be provided for to accomodate particular tastes and interests of students.

5 A different Conference Adviser will be assigned for each term, which will enable the group to become intimate with the modes of action and learning of a number of the faculty members during the junior and senior years. At times when special features may be discussed in the conference, the specialists for which our staff is notable will be invited to attend and take part in the mutual discussions, thereby giving opportunity for farther enlargement of the students' acquaintance and horizon of thought.

6 If you become interested in a particular investigation, even as early as the junior year, that would develop into a suitable thesis project, this may be substituted for the larger part of the usual laboratory work.

7 For the purpose of stimulating the group in the scholarly thoughtfulness regarding their careers that is needful for distinguished leadership in creative electrical engineering, near the end of the senior year each student in the group will present to his Conference Advisers an oral statement of his progress and accomplishments during the junior and senior years, with comments on his grasp of the electrical engineering field and its collaterals.

8 The foregoing seven numbered paragraphs relate particularly to the subjects of the junior and senior years which are taught by the electrical engineering department, but we are authorized by the faculty to extend to you like privileges for the other subjects in the curriculum of those years. If the group wishes to take advantage of this also, a second weekly conference will be provided, and a second Conference Adviser, for thesel atter subjects.

This plan will bring into the undergraduate work the fertility and individual interest usually thought of as only available in graduate work.

You will observe that the plan provides for what may be called an undergraduate seminar group, membership in which should be recognized as a privilege and honor; and it secures even more flexibility of program for gifted students than is now secured by the large degree of substitution available for the senior year.

This recognition of the exceptional student is not to be confined to the Massachusetts Institute of Technology, as an announcement from Dartmouth College indicates that a change in policy permits exemption of such a student from all regular class attendance, affording him such instruction as best fits his individual needs and freedom to pursue his work at whatever speed and to whatever distance his capacity permits. Surely these are encouraging signs of a progressive tendency in modern education.

An American Museum of Engineering

ON MONDAY, August 17, the presidents and the secretaries of the four National Engineering Societies met Samuel Insull at a conference luncheon given in his honor by John W. Lieb, the subject of the conference being the endeavor of founding a National Museum of Engineering and Industries. This movement, which started sometime ago, has beeen fortunate in recently securing the services of Mr. Insull as its leader. The following day Mr. Insull left for Europe for the purpose of making an intensive study of the more

important engineering and industrial museums abroad, and will return next month with plans well matured for the establishment of the proposed institution.

The committee appointed to consider the subject of engineering and industrial museums and to recommend what action, if any, the four National Engineering Societies should take toward their encouragement and support, recently submitted the following report in which they recommended—

(*a*) That there should be established in Washington, D. C., preferably under the administration of the Smithsonian Institution, an historical national museum to preserve the experimental and operating originals of epoch-making scientific, engineering, and industrial inventions and discoveries as a permanent record of great achievements.

(*b*) That there should be established in large centers of population of the United States, industrial museums of an educational type wherever local initiative manifests real interest in such undertakings and when substantial financial guarantees are assured for the foundation and endowment of such museums.

(*c*) That the engineering societies, because of their purpose and the character of their membership, should lend their encouragement and support toward the establishment of museums as set forth above.

(*d*) That the foundation of an historical engineering museum at Washington, as described in the above recommendation can best be achieved by substantial funds being raised by a private organization toward building and endowment, after which an organized appeal may be made to the Congress of the United States to establish and maintain such a museum with the aid of public funds as a departmental museum under the direction of the Smithsonian Institution.

(*e*) That wherever local initiative has been manifested and when substantial financial guarantees are assured for the foundation and endowment of industrial museums of an educational type in large centers of population, the Founder Engineering Societies should endeavor to secure the cooperation and support of their local organizations for such undertakings.

(*f*) That the Founder Engineering Societies should build upon the efforts which have already been made by the National Museum of Engineering and Industry by arranging for an adequate representation on the Board of Trustees of that organization. The organization named—incorporated under the laws of the District of Columbia and supported by a number of eminent engineers—has requested endorsement of its efforts by the Founder Engineering Societies, has requested nominations to its enlarged Board of Trustees from those Societies, and has created during the past few years public interest in the idea of an engineering museum in Washington.

(*g*) That in order to provide a practical basis for coöperation with the National Museum of Engineering and Industry on the lines noted above, it is desirable that the Founder Engineering Societies assist in obtaining moral and financial support.

OBJECTIVES

The effort to establish museums of engineering and industry is a public movement of the kind of many which the engineers of the United States have from time to time encouraged and supported through their professional organizations. The objectives of such institutions as have been developed in other industrial countries are, on the whole, in keeping with the purpose of American engineering, which has been acceptably expressed as "the science of controlling the forces and of utilizing the materials of nature for the benefit of man, and the art of organizing and directing human activities in connection therewith."

The need for such museums arises from the complexity of modern industrial civilization, which has become organized on a scale so vast as to reach and influence every human being. The imperative human needs for food, clothing, shelter, transportation, communication, leisure for the improvement of the mind, and opportunity for the growth of the spirit, can now only be satisfied under conditions of modern life through the uninterrupted and harmonious operation of an enormous organism of industrial activities.

The development of the factory system has brought a great change in regard to popular knowledge of industrial processes and methods. Where formerly these were open to the sight and understanding of every one, in many cases family activities, they are now hidden behind factory walls and have ceased to be the heritage of the younger generation. Yet some knowledge as to the nature and growth of the industries fundamental to our daily life is essential to a full comprehension of our political and social institutions, and of the civilization of today. Museums of engineering and industry can be made an essential means of securing and perpetuating such intelligent understanding.

Progress comes from the work of individuals and by the development of specific improvements. It is essential that the nature of these improvements should be widely understood in order that duplication of effort be reduced, possibilities of future advancement be indicated and invention stimulated. This essential can also be satisfied by museums of engineering and industry. By gathering together and exhibiting the originals of epoch-making scientific engineering and industrial inventions there will be preserved and made available for the inspiration of youth an imperishable record of great achievements and of the heritage which each generation received from its predecessor. Through working models illustrating natural laws there may be brought to many an understanding of the forces and methods that form the basis of modern industry. Through organized exhibits of industrial processes there may be shown the origin of the articles and utilities of every-day life, their methods of production, the contribution of human hands and brains in their output, and the part they play in establishing and maintaining standards of living.

The engineering profession has repeatedly presented the ideal of increased human well-being through making the benefits of human industry and invention available in larger and larger measure to all peoples, and has dedicated itself to the realization of that vision. The knowledge and understanding which engineering and industrial museums disseminate create a force in achieving this ideal, and may be made a mighty influence for the further improvement of industry and the establishment of industrial peace.

In view of these possibilities your committee unqualifiedly believes that the Founder Engineering Societies should assume and discharge a full measure of responsibility for the establishment of engineering and industrial museums, and to that end expresses its judgment in the preceding recommendations.

JOHN R. FREEMAN, *Chairman*
FRANCIS LEE STUART } for the American Society of Civil Engineers

A. S. DWIGHT for the American Institute of Mining and Metallurgical Engineers

L. P. ALFORD
C. R. RICHARDS, *Secretary* } for The American Society of Mechanical Engineers

F. J. SPRAGUE
J. P. JACKSON } for the American Institute of Electrical Engineers.

"Two museums of this kind exist and have existed for a number of years," writes Dr. M. I. Pupin in the September *Journal* of the A.I.E.E., "one at South Kensington, in London, and the other at Munich, Germany, and their importance cannot be overstated. He who knows the splendid influence which the Munich museum has exerted upon the development of German engineering and industry cannot help wondering why a similar national museum was not founded in the United States long ago. The project is large, but no project is too large provided its national importance is justified." Continuing, Dr. Pupin writes:

One cannot visit the American Museum of Natural History, of New York, without being deeply impressed by its educational value. In a national museum of engineering and industries one will have a splendid opportunity to speak to past generations of engineers and inventors and they will teach him. Science may be described as the study of nature's language and logic. The most concrete illustration of nature's language and logic are the structures which were created by the inventive faculty of man and developed by engineering and the industries. For instance, a properly arranged exhibit of power generators from Watt's steam engine up to the latest creations of gas engines speaks a more eloquent language than any orator ever employed, and shows a more convincing logic than one can find in any system of ancient or modern metaphysics. Address this language and this logic to the right kind of youth and watch for the results. The watching will be brief, because you will not have to wait very long for the active imagination of youth. Give the youth a chance to be thrilled by the achievements of past generations and do not rely upon books alone to produce this thrill. Put them in touch with the things which record these achievements and with the lives of men who made those things. That is the true educational mission of a national museum of engineering and industry, and its importance in our national life cannot be overestimated.

Course in Military Engineering at M.I.T.

BY MAKING use of one summer semester and a regular academic year of study, the Massachusetts Institute of Technology is offering this year for the first time a special course for army officers, leading to the degree of S.B. in Military Engineering. The following excerpt from the official announcement contains a statement of the purpose and scope of the course.

This course, which covers a full calendar year, has been prepared specifically to cover such engineering work as an army officer is called upon to carry out. In the summer preceding the academic year the men are given a review of Applied Mechanics, starting in at the beam theory and finishing with the theory of elasticity. At the same time the men are taking a course in Structures and a course in Heat Engineering covering thermodynamics of perfect gases, saturated and superheated vapors, refrigerating machines, air compressors, steam engines, steam turbines and heating and ventilation. This is followed in the academic year by a thorough course in Structures, by a course in Railway and Highway Engineering, a course in Reinforced Concrete Design in which abutments and concrete bridges are particularly discussed, simultaneously with a course in the laboratory on the testing of concrete, and then later by a course on Concrete Building Design and Specifications. Materials of Engineering, which treats of the specifications for engineering material, takes up the cooling curves of iron and metallography, this course being parallel with a laboratory course in Heat Treatment and a laboratory course on the behavior of metal under stress. In the electrical line the men are given the elements of Electrical Engineering, electrical engineering laboratory work, together with industrial applications of electrical power. The work in Heat Engineering is driven home by a laboratory course in the Engineering and Hydraulic Laboratory, this course consisting of 45 hours in the laboratory with 45 hours devoted to writing engineering reports. A course in Physical Chemistry, covering principally molecular structure, ties in with the course in Heat Treatment and Materials, and it is expected that this will leave the men in such shape that they will be able to understand intelligently X-ray work of steels and alloys.

and will understand the application of methods of photoelasticity in the determination of stresses in complicated pieces. A special course in General Bacteriology prepares the men for the course in Municipal Sanitation. A thesis, or original research of 135 hours' duration completes the requirements.

Industrial Engineering at Ohio State

BEGINNING with the academic year 1925–26 a course in industrial engineering is to be offered at Ohio State University, Columbus, Ohio, thus adding a new department to the College of Engineering.

John Younger, formerly chief engineer, truck division, Pierce Arrow Motor Car Company, and later during the war chief of the engineering division of the Motor Transport Corps, and publisher and editor of *Automotive Abstracts*, will be head of the new department.

In discussing his plans for the new department, Mr. Younger writes that the curriculum will present a general survey of what is meant by the term "industrial engineering." Other parts of the course will include studies of planning, so that a particular piece of work may have its necessary jigs, fixtures, and tools planned, its proper routing arranged, and a study made of the best way to produce it economically.

The students will be instructed in shopwork not so much in the design and operation of machine tools as in the proper selection of them for their economic advantages. Studies of personnel and factory accounting problems will also be introduced into the curriculum, and methods of analysis by means of statistics and curves will be taught.

New Haven Machine-Tool Exhibition

THE Fifth Annual Machine Tool Exhibition, held at Mason Laboratory of Yale University in New Haven, Conn., September 8 through 11, was an unqualified success. Fifty-nine exhibitors, representing one hundred and ten manufacturers, used all the space made available. Throughout the day the exhibit was comfortably filled, and in the evening it was crowded. The attendance was estimated at from 15,000 to 20,000. So much of an institution has this affair become within five years, that it has now been incorporated under the laws of Connecticut to insure its continuity and to remove the element of personal liability. Tentative plans have been made to hold the next exhibition September 7 through 10, 1926, and a large amount of space therein has already been signed for—an indication of the estimate of the industry.

The technical sessions under the auspices of the Machine Shop Practice Division, A.S.M.E., were unusually interesting and unusually well attended. The paper on Theory and Practice of Centerless Grinding by W. J. Peets, of the Singer Manufacturing Company, showed the wide application of this process and revealed it as not only one of high production but of great accuracy as well. A written discussion of this paper was submitted by W. W. Seabury, of the Ford Motor Company, and was illustrated by numerous successful applications of the process in the motor industry. The practical range of the system now includes diameters from $^1/_{16}$ in. to 3 in. and its adoption has multiplied output to a surprising degree.

The paper on Precise Cylindrical Lapping by Paul M. Mueller, of the Pratt & Whitney Company, was remarkable in the revelation of the degree of accuracy which is being achieved by the use of simple machinery. Mr. Mueller spoke freely in terms of the millionth of the inch, which with the help of the interferometer and the millionth comparator is now a practical unit of measurement to the commercial gage maker. A written discussion of the paper was submitted by C. T. Appleton, of the Reed-Prentice Company.

Luther D. Burlingame, of the Brown & Sharpe Manufacturing Company, read a paper on High-Speed Cutting of Brass in which he indicated that as yet there was little standardized practice in tools or coolant. The small diameter of much of this work and the high cutting speed employed upon soft materials have resulted in the modification of standard tools to give unusually high spindle speeds and coarse feeds. The rate of production on some of his samples was extraordinary. One of his exhibits was a vial of perfectly formed watch screws so small that a strong glass was required for their examination.

The evening meeting on Wednesday was one of the most interesting. Papers of vital interest to the machine-tool industry which were presented by O. B. Iles, President of the Machine Tool Builders Association, on The Future of the Machine Tool Industry, and by W. H. Rastall, Chief of the Industrial Machinery Division of the Department of Commerce in Washington, on Foreign Markets, have since been published at length in the trade press. William B. Stout's paper on All-Metal Airplanes was read and amplified by G. H. Hoppin, his close associate in the Airplane Division of the Ford Motor Company. This was illustrated by slides and moving pictures showing that commercial air transport of freight as well as passengers is now a reality in the United States.

The keynote of the session on Shop Training Methods was struck by J. P. Kottcamp, of the Johns-Manville Company, in his paper on Technical Training for Industry. His proposal of the more general adoption of the intensive two-year engineering course under rigid industrial standards provoked very active discussion among the many educators present. With Prof. Dugald C. Jackson as chairman and with the able discussers at his call, the meeting was lively and interesting through its full period.

A study of the exhibits confirmed the belief that developments since 1918 have rendered most wartime machine tools of the production variety definitely obsolete. At the same time it was evident that parallel progress has been made by most manufacturers in the same lines, so that their future volume of business will be largely dependent upon ingenuity of tooling, quality of tool work, and service given as production engineers as far as their particular machines are concerned. Further elimination of the human element was also evident in the more general application of automatic movements to such machines as drill presses and milling machines.

The Exhibition was unique in that it brought together two leading educational institutions, Yale University and the A.S.M.E., and a commercial body, the New Haven Chamber of Commerce, in an event of outstanding educational and industrial value.

Compensation for Engineering Work

COMPENSATION for engineering work has recently received attention in the columns of MECHANICAL ENGINEERING and aroused much comment. An interesting comparison of engineering compensation with legal compensation was made in the dissenting opinion given by Commissioner Eastman of the Interstate Commerce Commission on the decision of the I. C. C. as to the compensation that reorganization managers should receive in the reorganization in Missouri, Kansas and Texas Railroad. The following paragraphs of Mr. Eastman's statement are illuminating in this respect:

"The record leaves the impression that one of the most important, intricate, and difficult pieces of work in connection with the reorganization was the work done by the firm of engineers in reporting in detail upon every main and branch line of the entire property, covering not only physical conditions and valuation but methods of operation, past earnings, and future prospects. For this work, as aforesaid, a total of $156,675 was paid, and this covered not only the original report but a retainer throughout the period of the reorganization work. This amount may be contrasted with the $750,000 now sought by counsel for the reorganization managers.

"Undoubtedly the reorganization of this property involved a great many complexities, both financial and legal. It was a task requiring long and patient perserverance and a high order of experience and ability. The two banking houses possessed excellent standing and commanded confidence in the financial world, and the same may be said of their two firms of counsel. The work also involved some risk of loss of profits in the event of failure or partial failure of the plan. How are we to weigh the pros and cons and reach a conclusion as to the compensations which were fair and reasonable under the circumstances?

"It was testified in this case that the services of one of the lawyers were worth $500 per day. Men receiving such compensation usually entertain a righteous conviction that they correspond with intrinsic worth, although at the same time they may be incensed that painters or carpenters should assume to demand and be able to exact $15 per day for their services."

Book Reviews and Library Notes

THE Library is a coöperative activity of the A.S.C.E., the A.I.M.E., the A.S.M.E. and the A.I.E.E. It is administered by the United Engineering Society as a public reference library of engineering and the allied sciences. It contains 150,000 volumes and pamphlets and receives currently most of the important periodicals in its field. It is housed in the Engineering Societies Building, 29 West 39th St., New York, N. Y. In order to place its resources at the disposal of those unable to visit it in person, the Library is prepared to furnish lists of references to engineering subjects, copies of translations of articles, and similar assistance. Charges sufficient to cover the cost of this work are made.

The Library maintains a collection of modern technical books which may be rented by members residing in North America. A rental of five cents a day, plus transportation, is charged. In asking for information, letters should be made as definite as possible, so that the investigator may understand clearly what is desired.

Diesel and Oil Engineering Handbook

DIESEL AND OIL ENGINEERING HANDBOOK. By Julius Rosbloom. The Technical Publishing Co., Los Angeles, Cal., 1925, third enlarged edition. Fabrikoid, 5 × 7 in., 794 pp., illus., diagrams, tables.

THE book is of interest if only in showing that the Diesel and oil engineering industry have reached such a high stage of commercial importance as to warrant an extensive handbook of this character.

This is all the more so as the author wisely presumes apparently that any one having this book at hand would also have at his disposal one of the standard engineering handbooks. He has, therefore, omitted entirely such material as can be found in them and confines himself to materials of specific interest to Diesel and oil-power users generally. The book covers the following subjects and is well illustrated: General discussion of leading types of Diesel engines for land and marine services; detail description of various types of Diesel machinery, operating instructions, emergency repairs, etc.; Diesel electrical drives, electrical equipment, electrical auxiliaries, questions and answers, etc.; technical terms and formulas for license laws, questions and answers, indicator practice, etc.; treatise on fuels and lubricants, tables, oil-purification methods, lubrication devices, fire-prevention systems, questions and answers; service pumps of different varieties, telemotors, compressors, tables, engine-room piping, etc.; general discussion on various types of heavy crude-oil engines of Diesel principle and semi-Diesel classification; rules for licensing of engineers on motor ships on American and British marine services, maneuvering signals, etc.; machinery equipment in the petroleum industry, tables, etc.; latest developments in the field of oil engineering, such as Diesel-electric locomotive, Diesel automobiles, Diesel-portable machinery for various industrial usages, cooling towers.

Transactions of First World Power Conference

THE TRANSACTIONS OF THE FIRST WORLD POWER CONFERENCE, LONDON, JUNE 30 to JULY 12, 1924. London, Percy Lund, Humphries & Co., Ltd., 1925. Cloth, 10 × 6 in., 4 vols. 6423 pp., illus., maps.

These four volumes contain the papers presented to the first World Power Conference, and the discussions that they provoked. A fifth volume, containing an elaborate index, is in preparation.

Few engineering congresses have left so extensive and elaborate a record of their proceedings; and few have excited such widespread interest, if the use of this report may be taken as an indication. The Conference brought together delegates of many countries, as well as a large number of other engineers from all quarters of the globe; the papers were of high quality, so that the published record is a valuable summary of power resources and their present utilization.

The scope of the Conference was a wide one. The available and utilized power resources of the world were reported upon very fully, the papers upon this subject filling an entire volume. Upon the production of water power, the preparation of fuels and the production of steam power, the contributions were sufficient to fill another volume which is a comprehensive review of the best practice of various lands in the development of water powers, water-wheel design, the distillation of coal, producer gas, peat, shale oil, and in the utilization of wood waste, as well as of modern steam power plant design and practice.

The subjects contained in volume three deal with internal-combustion engines, gas, such sources of power as wind, alcohol, and natural steam, the transmission and distribution of power by electricity, research, standardization, and illumination.

Papers on a variety of subjects appear in volume four. The varied uses of power in industry and for domestic purposes, in electrochemistry and electrometallurgy, and for transport on land and sea are discussed and described. Various economic and financial problems are treated, as are the governmental policies of various countries. Several papers on education, health, and publicity conclude the work.

A work so broad in scope and so varied in character cannot fail to offer something of interest to every engineer, whether he deals with the production of power or its use. The list of contributors includes many of our best authorities and the articles, with their numerous illustrations and maps, give a graphic picture of the present state of power engineering.

H. W. C.

Books Received in the Library

BAU UND BETRIEB MODERNER KONVERTERSTAHLWERKE UND KLEINBESSEMEREIEN. By Hubert Hermanns. Wilhelm Knapp, Halle A. Saale, 1925. Paper, 7 × 10 in., 251 pp., illus., diagrams, 12 gold marks.

A detailed account of modern practice in the manufacture of bessemer steel. The book opens with a comparison of the open-hearth and bessemer processes and of the economic advantages of each. The chemical bases of the bessemer process and the heat balance in the converter are then explained, after which the book takes up successively the location and arrangement of large bessemer works. The details of the necessary equipment are then discussed. The last three chapters are devoted to dolomite brick, slag-grinding plants, and small bessemer works.

FACTORY ORGANIZATION AND ADMINISTRATION. By Hugo Diemer. Fourth edition. McGraw-Hill Book Co., New York, 1925. Cloth, 6 × 9 in., 298 pp., illus., forms, $4.

This well-known book on the organization and administration of factories has been revised in the light of the development of practice in production, factory organization and the science of management, so that it is adapted in the new edition to our present practice. It is intended for all those, officials, accountants, superintendents, and students, who wish to understand the problems that arise and the methods of solving them.

GRUNDLAGEN UND NEUERE FORTSCHRITTE DER ZAHNRAD-ERZEUGUNG. By Karl Kutzbach. V. D. I. Verlag, Berlin, 1925. Paper, 7 × 10 in., 70 pp., illus., diagrams, tables, 5 marks.

Professor Kutzbach's pamphlet discusses the fundamentals of the whole field of the generation of gearing in a practical way, suited to the needs of students and designers. Recent processes of gear making are described, and a comprehensive view of present practice in design and manufacture is given. There are appendices explaining the usual notation and on the correction of tooth profiles.

HYDRAULICS. By R. L. Daugherty. Third edition. McGraw-Hill, New York, 1925. Cloth, 6 × 9 in., illus., diagrams. $3.00.

This edition has been entirely rewritten to bring it up to date and to improve the arrangement and the presentation of the sub-

ject. The book is intended as a text for students who must cover a wide field in hydraulics in a limited time, therefore attention is given mostly to matters of fundamental importance and but little space is given to things that are of small practical value. By this presentation of fundamental principles, a basis for specialization in certain topics is laid.

AN INTRODUCTION TO FLUID MOTION. By W. N. Bond. London, Arnold, 1925. Cloth, 94 pp., diagrams, 5 × 7 in., $5.00.

Mr. Bond is of the opinion that the study of fluid motion has advanced to a point where it should be recognized as an independent branch of physics rather than as a branch of mathematics (hydrodynamics) and as hydraulics, a branch of engineering. His book aims to further this view by providing an introduction to the subject for those undertaking advanced study. It considers briefly the various properties of the fluid, gives references to the more important original publications and describes modern experimental work.

MÉCANIQUE DE L'AVIATION. By Lieut-Col. Alayrac. Gauthier-Villars et Cie., Paris, 1925. Paper, 6 × 10 in., 350 pp., diagrams, 50 fr.

Aims to provide a course in mechanics which will equip the aeronautical engineer with general methods which will enable him to solve the questions that arise and free him from reliance upon general formulas of uncertain merit. The first portion of the book studies the exterior forces acting upon an airship. The author then proceeds to an analysis, by classic methods, of the movement in space of the center of gravity of the airship, from which he proceeds to a study of the movement of the airship about its center of gravity and of the problem of stability. A final chapter discusses navigation and the radius of action.

NATURAL GASOLINE; Testing, Manufacturing and Properties. By G. G. Oberfell and R. C. Alden. Published by the Authors, Tulsa, Okla., 1924. Fabrikoid, 6 × 9 in., 533 pp., illus., tables, graphs, $7.50.

The authors of this work have collected a large mass of data upon the testing, manufacture and properties of natural gasoline, and arranged it in a form convenient for those engaged in the industry. The first section, on testing, gives methods for testing gas, gasoline and absorbent oils. In section two, on manufacture, the application of the fundamental physical laws is explained, and the design and operation of recovery plants, both oil and charcoal, are explained. The physical and chemical properties of natural gasoline are given in section three. Section four contains a miscellany of useful tables and graphs, and a bibliography of books, articles, and patents.

OIL REFINERY SPECIFICATIONS. By A. L. Nugey. Chemical Publishing Co., Easton, Pa., 1924. Cloth, 6 × 9 in., 210 pp., diagrams, tables, $3.50.

The author gives specifications for the buildings, equipment, and machinery used in American oil refineries of various types, accompanied by brief accounts of the refining processes in use and by practical advice on many details of construction. Specifications are also given for filling stations and shipping containers. The book is the first of its kind, the author states, and is based on long personal experience in the design and construction of refineries.

RESISTANCE OF EXPRESS TRAINS. By C. F. Dendy Marshall. Railway Engineer, London, 1925. Cloth, 9 × 12 in., 76 pp., illus., tables, $1.

Although a great deal of experimental work upon the important subject of train resistance has been done, there has never been a book in English dealing with the steam-railroad side of it. Mr. Marshall here gathers up the results, hitherto scattered through many publications, and attempts to throw new light on the subject. Much of his work is the result of studying the available records of experiments with trains in the light of aeronautical science. He analyzes the effect of winds upon the resistance more closely than has been attempted previously and makes definite proposals which he believes would lead to a reduction of resistance, to smoother running, and to economy.

RESISTANCE OF MATERIALS. By Fred B. Seely. John Wiley & Sons, New York, 1925. Cloth, 6 × 9 in., 442 pp., diagrams, tables, $3.75.

The author of this textbook is professor of theoretical and applied mechanics at the University of Illinois, and the book, it may be assumed, represents the general course given at that school. The first, and longer, part of the work dealing with the mechanics of materials, treats chiefly of the application of the principles of analytical mechanics and of the experimental laws of structural materials to the analysis of the members used in structures and machines. The second part treats of the force-resisting properties of engineering materials. In part one, rational methods are developed for the design of the common types of force-resisting members used in engineering structures. Part two investigates those properties of materials from which the suitability of the material for various structural uses may be determined and considers the tests that will measure these properties.

REWINDING SMALL MOTORS. By Daniel H. Braymer and A. C. Roe. McGraw-Hill Book Co., New York, 1925. Cloth, 6 × 9 in., 247 pp., illus., diagrams, charts, $2.50.

The motors to which this book refers include all those in common use for portable drills, grinders, automobile starters, sewing machines, desk fans, vacuum cleaners, washing machines, and similar shop and dwelling equipment. The authors have attempted to compile details about the procedure in winding these motors which will enable an experienced winder to rewind or change them without difficulty.

RIGID AIRSHIP: a Treatise on the Design and Performance. By E. H. Lewitt. Isaac Pitman & Sons, London and New York, 1925. Cloth, 6 × 9 in., 283 pp., illus., plates, diagrams, $8.50.

The first complete work on the subject, the author asserts, to be published in the English language. The author writes on the basis of personal experience and intends the work for designers and students of aeronautics with good engineering training. The field is restricted to structural design and performance; engines and machinery are not discussed.

SHALE OIL. By Ralph H. McKee. Chemical Catalog Co., N. Y., 1925. American Chemical Society. Monograph series. Cloth, 6 × 9 in., 326 pp., illus., tables, diagrams, $4.50.

A summary of the shale-oil situation by a number of authors. There are chapters on the present state of the industry, on the origin, geology, and distribution of oil shales, on kerogen, on the analysis and evaluation of oil shales, on refining, and the nitrogen constituents of the oils, on the economics of the industry, and on American experimental distillation plants. A useful collection of abstracts is included, summarizing over eleven hundred books and articles, and there is a list of British and American patents.

SHEET METAL WELDING BY THE OXY-ACETYLENE PROCESS. Compiled by T. C. Fetherston. Linde Air Products Co., New York, 1925. Cloth, 6 × 9 in., 259 pp., illus.

Intended primarily for those working with light sheet metal. Gives a detailed description of practical methods for preparing and assembling work for welding, of the jigs and fixtures used, and of methods of welding, testing, and finishing. The applications of welding in the manufacture of utensils, metal furniture, barrels, automobiles, and various sheet-metal articles are illustrated, and there are chapters on shop equipment, organization, and cost records.

TECHNISCHES WORTERBUCH. . . . DES MASCHINEN UND SCHIFFBAUES. By Erich Krebs. Walter de Gruyter & Co., Berlin and Leipzig, 1925. Cloth, 4 × 6 in., 149 pp. 1.25 R.M. (.30).

A pocket-size dictionary of German words used in mechanical and marine engineering, with their English equivalents. Contains the terms used most frequently in connection with steam and internal-combustion engines, automobiles, pumps, hoisting machinery, machine tools, shipbuilding, and dredging. Printed in clear, legible type.

Erratum

ON PAGE 709 of the September issue of MECHANICAL ENGINEERING in the paper by L. D. Burlingame on High-Speed Cutting of Brass and Other Soft Metals on Standard Machine Tools, the time indicated under example 4 should be 5 seconds, not 7.